BULLETIN 632 AUGUST, 1942

Refrigerated Farm Apple Storages

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REFRIGERATED FARM APPLE STORAGES

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INTRODUCTION

Within the past 10 years there has been a steady increase in the number of fruitgrowers purchasing refrigeration equipment for their apples. Approximately 15 per cent of the more than 200 farm apple storages in this State are equipped with refrigeration units. Most of the refrigerated storages have been built during the past 5 years, and storage space on farms has increased at least 30 per cent within this period. During 1940 the increase was approximately 10 per cent. Since Ohio trade is inclined to buy fresh-packed fruit direct from the farm storage, there is a decided trend to continue the development of farm storages with as much refrigeration as can be arranged on a practical basis. Several states (1)[1] report similar increases in storage capacity on the farm and a larger percentage being refrigerated each year.

There are several explanations for this trend from the commercial warehouse storage to the farmer-owned refrigerated types. Growers catering to local or near-by markets find it an advantage to have ample supplies of good apples of several varieties to supply truckers and other buyers. There is a more orderly marketing of fruits under such circumstances. The fruit may come out of storage in much better condition, and high quality favors an increase in repeat sales. The grower with his own storage can perform his harvesting operations with a much smaller crew than is necessary for one who picks and packs for immediate shipment. During 1926 in Kentucky, when apples were selling for 25 cents per bushel at harvest time, many common storages were hastily constructed. Apples held until February and early March of the following year averaged 65 cents per bushel tree-run. Storage building cost amounted to about 32 cents a bushel. A similar marketing condition has prevailed during 4 of the past 12 years. As one expressed it, fruitgrowers more than paid for the storage the first year and had the house left. Excellent condition of the fruit at the end of the storage period is due, growers report, to their personal handling of the fruit and their care in providing proper ventilation and maintaining correct temperature. New York State (1) reports that the development of automatic cold storage features and the high prices offered late in the spring for McIntosh which have been kept in good condition have both given emphasis to the movement toward farmer-owned storages there. All growers seem to agree that the farm storage has proved beneficial in their marketing program and has helped solve some marketing problems. Arnold (2) of New York State, upon interviewing more than 300 storage owners in five states and in Canada, found the three most frequent reasons given by the operators for owning their storages to be: The cost of storage on the farm is less than that of commercial storage. The grower is better able to choose his market. The necessity of grading and packing at picking time is eliminated. Of the common storage operators, 61 per cent reported that they were interested in changing to refrigerated storages.

[1]Numbers in parentheses refer to Bibliography.

STORAGE AND REFRIGERATION COSTS

Although fruitgrowers might prefer their own storages, the economy of such an investment is of primary importance. A storage must pay its owner a return on his investment, as is expected of any piece of machinery or equipment. Usually the total investment, particularly the capital outlay made at the time of the construction of the building and the installation of the refrigeration equipment, is relatively large and represents a lien on the future income of the orchard business.

Several workers (1, 2, 13, 18) have shown that apples held in storage frequently bring a better price than those sold at storing time. This condition would naturally vary with the quantity of apples produced, their market value before and after storage, and storage costs. Past experience has proved that the price of apples rises during the winter and spring months and that the rise will usually justify storage costs and frequently return a profit on the storing in Ohio where roadside markets are available and a great portion of the farm apple trade is retail. Each individual grower may determine the value of a storage investment to himself by examining the records of prices received for fruit from month to month in his markets and comparing the increase in prices with the cost of storage.

In table 1 are given an example and a method of calculating very closely the average cost of storing fruit under refrigeration at the farm. Although the itemized cost in this table is based on 10,000-bushel capacity, the percentage of the total investment for each item of annual cost can be applied to refrigeration and building costs for any size of storage. The estimated costs in table 1 are above the average for Ohio. Many storages pay only 2 cents per kilowatt for current and consume 1½ kilowatts per bushel per season; however, their total cost for power will be the same. Twenty-five per cent of the total cost has been allotted to refrigeration, and this percentage may also be considered high. The depreciation on equipment is calculated on a complete obsolescence in 14½ years, which is a conservative estimate for the refrigeration equipment, but 5 per cent may be somewhat high for a good building. As the equipment depreciates yearly, it is necessary to arrive at a figure for interest on investment which will represent the average interest over a period of years, and 5 per cent is selected as being approximately correct under present earning rates for money.

TABLE 1.—Estimated cost of operating a cold storage on the farm*

Itemized costs based on a storage of 10,000-bushel capacity

Total cost: refrigeration $2,500; building $7,500

Items of annual cost	Per cent of the total investment for—		Refrigeration	Building	Total
	Refrigeration	Building			
Depreciation on refrigeration equipment and building	7½	5	$187.50	$375.00	$562.50
Interest on depreciating refrigerating equipment and building	5	5	125.00	375.00	500.00
Tax and insurance	1½	1½	37.50	112.50	150.00
Maintenance	3	1	75.00	75.00	150.00
Power, 1 kilowatt per bushel at 3 cents	....	....	300.00	....	300.00
Total annual cost	....	....	725.00	937.50	1,662.50
Total annual cost per bushel	....	....	.07	.09	.17

*Figures on percentage of the total investment for equipment and for building are those suggested by companies installing farm refrigeration systems.

An indication of the reliability of the figures in table 1 can be gained from surveys of actual storage costs. Arnold (2) calculated the cost per bushel of 23 different storages, using 2½ per cent for insurance, 5 per cent for depreciation, 5 per cent for interest, 3 per cent for repairs, and 3.6 cents per bushel for power. He also included 1 cent per bushel for handling costs, which are not included in table 1. The storages included those with capacities of 5,200 to 179,200 bushels and total costs of $3,000 to $55,000. The costs per bushel stored ranged from 7 to 30 cents and averaged 16 cents, close to the figure given in table 1.

The 16- to 17-cent cost per bushel assumed to be a fair estimate of the cost to store and refrigerate apples at the farm or in cooperative storages is considerably below the charge for a similar service in commercial storages (22 to 25 cents per bushel per season). Some attention and risks are assumed by commercial storage operators for the growers' fruit, but the cost of this service would be less than transportation and handling costs to and from the city storage. The higher cost for storage in commercial warehouses is probably due to the greater overhead and some profit shown by this type of enterprise. There seems to be little doubt that home storage of fruits offers advantages and economies that can hardly be overlooked by the progressive fruitgrower.

COST OF BUILDING AND REFRIGERATION

Although a storage with refrigeration may be a good investment, the problem of the first cost is a serious one for many fruitgrowers. The only methods by which the burden of heavy cash payments can be reduced are long-time financing, when possible, and constructing the storage in various stages. The former method adds to storage costs and is to be discouraged unless the additional cost is justified through savings to be made by the investment. The second method offers possibilities, since such portions as insulated floors, refrigeration, and packing and grading rooms can be added at any time after the storage portion of the building has been placed in service. Frequently it is possible to construct a portion of the storage contemplated complete in one room and later add another room. It is also feasible to install a cooling coil of ample capacity for all future requirements but operate it with a small compressor which can be duplicated when the cooling load of additional storage space is added. Some storages are constructed of tile with no insulation to serve as common or air-cooled storages and later insulated.

The original cash outlay for a storage and equipment will vary considerably, depending upon the type of building, size, kind of insulation, and refrigeration equipment installed. Well-constructed common storages including dry storage, and grading, packing, and sales rooms should cost between 35 and 40 cents per bushel of capacity; the maximum should be 50 cents. Some of the larger buildings cost as low as 20 cents. The annual cost of operation per bushel, including depreciation, interest on investment, taxes, maintenance, and power, is close to one-tenth the total costs given. Refrigerated storages cost approximately twice as much as common storages. The increased cost is due to their superior construction and insulation, as well as the cost for refrigeration. Some have been constructed for less than the better air-cooled storages (as low as 16 cents per bushel), and some cost as much as $1.50, but 50 cents to $1.00 per bushel would include the majority, with a safe figure closer to the higher cost. The cost for electric power and items of depreciation and maintenance on refrigeration equipment increases the total annual cost of these storages from one-tenth to one-fifth the total initial cost per bushel. It is well to keep in mind, however, that costs do not necessarily bear any direct relation to the functioning of the storage, for some of the least expensive structures have kept fruit in excellent condition. Methods of safely lowering construction and refrigeration costs are discussed later in this bulletin.

COST OF REFRIGERATING A COMMON STORAGE

Frequently growers wish to know the approximate cost of refrigeration alone, since they may want to install such equipment in an existing storage or in a new building at a later date. The cost of such equipment varies considerably, just as does the cost of the building, and for the same reasons. For the purpose of estimation, refrigeration can be considered to cost from one-third to one-half the total refrigerated storage cost. For the larger storages, the refrigeration costs proportionately less, and the first figure applies. Thus, installed cooling equipment will range from 25 to 50 cents per bushel of capacity for the smaller storages and from 16 to 33 cents for the larger storages (over 10,000-bushel capacity). The mean or intermediate values should apply in most cases. The cost of refrigeration alone per bushel per season is greater than one-fifth the total seasonal cost, since the item of power does not apply to building costs. An approximate figure would be one-half the total seasonal cost per bushel, or from 5 to 10 cents per bushel per season. These figures are based on the author's observations and are presented only to give an approximate idea of costs, which vary widely with materials and equipment and the location of the storage.

A second method of estimating the initial cost for refrigeration installed ready to operate would be to base it on the approximate "tonnage" required. Such "tonnage", as explained later, and given for various sizes of storages in table 14, should cost about $500 per ton. The cost to own and operate refrigeration, although somewhat variable depending upon equipment and total time of operation, will be close to $120 per ton per season, based on a total operation of 120 days, or approximately $1.00 per ton of refrigeration delivered, since the rating of refrigeration machinery is based on its delivery of tons of refrigeration per 24 hours of running time, as explained later.

THE STORAGE BUILDING

The construction of common storages has been discussed in Ohio Bulletin 573 (8). Much of this information applies to the refrigerated storage as well, and to eliminate duplication, the present discussion will be confined to the pertinent facts relating to construction where refrigeration is to be installed at the time of building or at a later date.

LOCATION

The location of the storage building is usually dependent upon the home and orchard site. Nearness to the road and home is important, since the storage is the center of selling, grading, and packaging operations during the winter months. Close attention to its operation is imperative for uniformly good results, and when sales are made from the storage or an adjoining salesroom, it saves many steps to have the stored apples close to the house and to the road. It is also of value to keep the storage room, in particular, well shaded from the sun and protected from prevailing winds. Protection may be most easily accomplished by judicious plantings. Occasionally it is possible to choose a site in a ravine or other low spot where cold air drainage will tend to maintain lower air temperatures than occur on surrounding elevated areas.

Choosing a site where a portion or all of the building can be banked with earth or actually underground is not recommended, since as much insulation must be used as when the structure is entirely aboveground, and during the colder months of the storage season, greater quantities of heat will pass into the storage from the relatively warm soil than from the colder air. The soil is cooler than the air during the early portion of the storage period, but during the major portion, the air temperature is lower than the soil temperature (50 to 60° F.), and there is less heat moving into the storage to absorb on the cooling coils when the walls and roof are exposed to the air. The temperature is fairly constant beneath a storage floor and may be taken as 50° to 60° F. throughout the year. The same thickness of insulation is used for either an above- or belowground storage, since this material is installed to reduce heat leakage into the storage room to a minimum during periods of high outside air temperatures, and nothing would be saved by reducing the thickness of insulation below the ground level, since more, rather than less, insulation would be economical during the period when heat continues to flow into the underground room while an aboveground room may actually be losing heat to the colder outdoor air. The savings in refrigeration will more than equal the cost of moving a few inches of soil away from the foundation of a structure built into a bank.

Drainage is very important in choosing a site, for if free water cannot be kept from under or around the building by artificial drainage methods, it is necessary to locate the structure where natural drainage is adequate.

SIZE AND SHAPE

The size of a storage structure is entirely dependent upon the desired capacity and the services to be accommodated. It is always most economical to include such areas as grading, packing, loading and unloading, dry storage, sales, machine storage, workroom, and machine room within one structure unless any of these service areas can be constructed of less expensive materials elsewhere, because with this plan, many walls can be common to two rooms, and total wall area can be reduced. Raising the roof may provide ample dry storage for containers at a very low cost. Closing in a portion of a roofed-over loading platform will supply office or sales room economically. The incorporation of such service areas should be planned before, not after, building has started.

The storage room should be planned for the eventual needs of a growing orchard, and, in addition, 75 to 100 per cent of the storage room area should be provided for grading, packing, temporary storage, handling, and loading. The average yield of the orchard at full bearing age should determine the size of the storage, since it is not economical to make provision for the very heavy crops produced occasionally. Building so that the storage can be enlarged to meet later demands is economical and conserving of limited finances.

A close estimate of the size of the storage room can be made by allowing 2½ cubic feet of space per bushel to be stored. This allowance will provide space for aisleways and ample headroom for air circulation by means of open fans or ducts.

Large storages are more economical of space, since more bushels of fruit can be stored per square foot of walls, floor, and ceiling, as shown in the last four columns of table 2. The importance of this relation is not the lower wall cost per bushel alone, but also the lower heat leakage per bushel through the walls and, therefore, the lower power cost for refrigeration per bushel of fruit cooled. When the size of a storage is doubled in all dimensions, there is only one-half the surface area per unit volume to lose or gain heat. The larger the storage, the more bushels can be stored per square foot of insulation required.

The shape of the storage, particularly the insulated portion, also affects the economics of construction and operation. To secure the maximum capacity with the least total area of walls, floor, and ceiling, the cube or equal-dimension building should be constructed. Ceiling heights are limited by the practical height to which crates can be stacked, with 14 feet a maximum. Very wide buildings require more columns for roof support or more elaborate and costly self-supporting trusses. For these reasons, the most common ceiling heights are 12 feet, room spans, 40 feet, but whenever possible, higher and wider spaces should be used. In table 2 are listed two or more possible dimensions for each of six storages with 5,000- to 20,000-bushel capacities. The floor area of each group of storages is nearly the same, but the total wall and total superficial areas are somewhat different, depending upon the size and length-to-width ratios.

The figures in table 2 indicate the size of areas requiring varying thicknesses of insulation and provide a rapid method of estimating material costs for storages of various sizes. These figures also indicate the proportion of the total heat leakage area ascribed to the floor, where insulation is often neglected because of its greater cost per square foot. The differences between storages of various shapes and sizes would be greater if the room height could be increased to 14 feet.

In the last four columns (table 2) are given the ratios of the storage capacity in bushels to the various areas, such as wall, floor, or ceiling, and combinations of the three, including the total superficial area. Since the capacities were based on 2½ cubic feet per bushel, these ratios are also directly related to the cubical content of the buildings. Since the height was fixed in each case at 12 feet, the storage space per square foot of floor or ceiling remains constant, and any saving is made in total wall area. This saving may amount to more than 3 bushels per square foot in the largest over the smallest storage included in the table.

The greatest savings in insulation and building materials can be made when the ceiling height is increased. For instance, the storage capacity of a 40 by 60-foot building can be increased equally by a 2-foot rise in height or by a 12-foot extension in length. The latter method would require 160 square feet less additional wall but 480 square feet more ceiling, floor, and roof area than the former, and the costs, including those for insulation, would be proportionally greater. It is frequently possible in existing buildings to excavate the floor to a lower level with no weakening of the foundations or rest of the structure and, thereby, increase the capacity of the storage room several hundred bushels (one or more layers of containers) at a very moderate cost.

TABLE 2.—Storage sizes as related to capacity and to insulation requirements

Approximate capacity	Approximate capacity	Dimensions	Floor area	Wall area	Floor and ceiling area	Total superficial area	Ratio of storage capacity in bushels to— Total superficial area	Wall area	Ceiling or floor area	Wall and ceiling or floor area
Bushels	*Cubic feet*	*Square feet*	*Square feet*	*Square feet*	*Square feet*	*Square feet*	*Square feet*	*Square feet*	*Square feet*	*Square feet*
5,000	12,500	25½ × 40 × 12	1,020	1,572	2,040	3,612	1.38	3.18	4.90	1.92
		32 × 32 × 12	1,024	1,536	2,048	3,584	1.39	3.26	4.88	1.95
7,500	18,750	32 × 50 × 12	1,600	1,968	3,200	5,168	1.45	3.81	4.69	2.10
		40 × 40 × 12	1,600	1,920	3,200	5,120	1.47	3.90	4.69	2.10
10,000	25,000	32⅔ × 62 × 12	2,026	2,272	4,052	6,324	1.58	4.40	4.93	2.32
		40 × 50½ × 12	2,020	2,172	4,040	6,212	1.61	4.60	4.95	2.38
		45 × 45 × 12	2,025	2,160	4,050	6,210	1.61	4.63	4.93	2.39
12,500	31,250	36 × 72 × 12	2,592	2,592	5,184	7,776	1.61	4.82	4.82	2.41
		40 × 65 × 12	2,600	2,520	5,200	7,720	1.62	4.96	4.80	2.44
		51 × 51 × 12	2,601	2,448	5,202	7,650	1.63	5.11	4.80	2.49
15,000	37,500	38 × 82½ × 12	3,135	2,892	6,270	9,162	1.64	5.19	2.79	2.49
		40 × 78½ × 12	3,140	2,844	6,280	9,124	1.64	5.27	4.80	2 50
		56 × 56 × 12	3,136	2,688	6,272	8,960	1.67	5.58	4.78	2.57
20,000	50,000	40 × 102½ × 12	4,100	3,420	8,200	11,620	1.72	5.84	4.87	2.66
		64 × 64 × 12	4,096	3,072	8,192	11,264	1.77	6.51	4.87	2.79

INSULATION AND OTHER MATERIALS

A discussion of insulation and other materials used in the construction of storages will be found in Ohio Bulletin 573 (8). Lumber is usually less expensive, initially, than masonry, including tile, especially when local lumber, rough-sawed, is available. In addition, more farm labor for a given material cost can be employed on lumber than on tile, brick, or cement blocks. The latter materials are preferred to lumber, however, because of their resistance to deterioration from all causes and to fire.

Loose-fill materials, such as ground cork (several grades depending on fineness), shredded redwood bark, and spun glass are preferred because of their low cost per unit of thermal resistance as compared with the board-type materials, whose first cost and installation cost are relatively high. The loose-fill type of wall is economical even when the additional cost of a double masonry, tile, or wood wall is included in the finished insulation costs. The usual procedure with masonry construction is to have an 8- or 12-inch outer wall with a 4-inch inner wall. The outer wall or both walls serve as bearing surface for trusses, beams, ceiling joists, and roof. For a wood structure, 2 by 6 or 2 by 8 studs are erected, and sheeting of some kind is placed on both sides.

A relatively new material, completely inert and fireproof, which shows practically no deterioration under any conditions, is a building block made of more or less cellular material having a core filled with the same material. Blast furnace slag is the raw material used in constructing the block and as a loose-fill insulation. As an example of its insulation value, the following comparison can be made: An 8-inch wall of concrete composed of sand, gravel, and crushed stone has a cold and heat transmission factor (U) of 0.65.[2] Common concrete blocks made of this material are slightly more insulating because of the cores or voids in the block, and the factor is reduced to 0.55. The thermal transmission factor of blocks made of the blast furnace slag is only 0.25, less than half that for the concrete block. Filling the voids or cores in the blocks with the basic slag in loose-fill form still further reduces the thermal coefficient, to 0.108, which is equivalent to that of a wall with 3 inches of corkboard or its equivalent and is the minimum requirement for an apple storage wall, ceiling, or floor. Several commercial concerns license local cement block manufacturers to construct blocks of this material.[3] It is also feasible to use the blast furnace slag for home-mixed floors or walls and obtain considerable insulating value.

There is another material which deserves wider use in apple storage construction, zinc-coated metal sheets, commonly known as galvanized sheeting, which have considerable durability and strength for use under the high humidities in apple houses.[4] The fire resistance of this material and protection from lightning when grounded make its use economical on roofs and sidewalls, as well as for interiors (23). The control of rats and mice is better when the storage is lined with sheet metal or wire cloth (fig. 1). Condensation can do

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[2]The transmission factor "U" is expressed in B. T. U. per hour per square foot of wall per degree Fahrenheit of temperature difference between the inside and outside air. The outside air velocity is taken at 15 miles per hour. One B. T. U., or British thermal unit, is the amount of heat required to raise the temperature of a pound of water 1 degree Fahrenheit.

[3]A few companies specializing in insulating blocks and loose-fill aggregates are: The Waylite Company, 105 West Madison St., Chicago, Ill.; The Celotex Corporation, 919 North Michigan Ave., Chicago, Ill ; The American Aggregate Company, 1002 Walnut St., Kansas City, Mo.; National Cinder Concrete Products Assoc., 1600 Arch St., Philadelphia, Pa.; Munn and Steele, Inc., 130 Lister Ave., Newark, N. J.

[4]American Zinc Institute, 60 East 42nd St., New York City, N. Y.

no harm when it does occur on galvanized metal surfaces. Asphalt, or, better still, zinc, paints can be used to protect such materials from moisture and the weather if rust spots appear. Insulation can be placed on, or adjacent to, zinc-coated metals, for this material is an excellent vapor seal, particularly if the joints are lapped and sealed by soldering or made fairly tight by caulking with asphalt or other waterproof material covered with vaporproof paint. Grades of galvanized metal at least as heavy as 28 gauge and coated with 2 ounces of zinc per square foot are recommended, since their longer life more than offsets their greater cost.

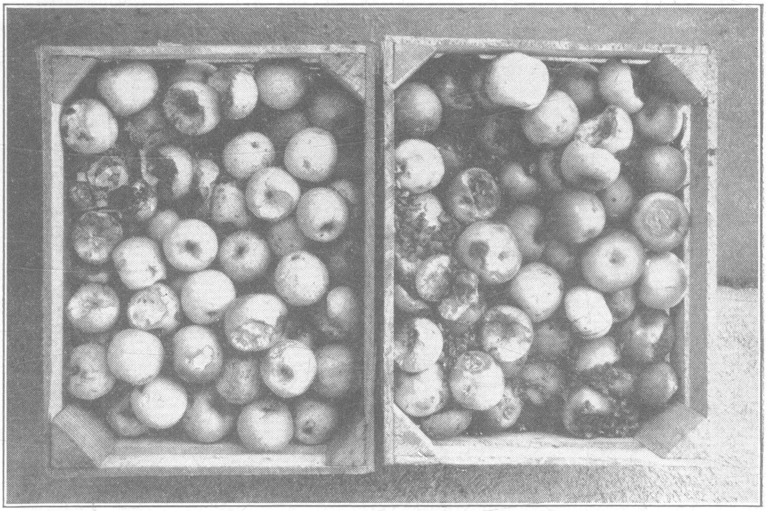

Fig. 1.—Apples damaged by rats in storage. This damage could have been prevented by the use of wire mesh screen or sheet metal.

A rat cannot climb above an 8-inch strip of metal, which will keep rats out as well as a complete covering of wire mesh screen.

THICKNESS OF INSULATION

The proper thickness of insulation for apple storage walls, ceiling, and floor is the economical thickness, for all heat coming through the insulation must be absorbed by the cooling unit at a certain cost per ton of refrigeration. The cost of insulation should be based on the cost per unit of thermal resistance installed. Balancing the cost of refrigeration employed in removing the heat coming through the insulated surfaces against the cost of the insulation used to reduce the flow of heat is one approach to the problem. Accordingly, table 3 was prepared. By solving the equation $N=\sqrt{\frac{F_1 \times TD}{F_2}}$[5] for any values of F_1, F_2, and TD determined by entering table 3 with a given thermal coefficient (K)[6] for a storage wall, ceiling, or floor, cost of insulation per board foot installed, cost of refrigeration per ton, and temperature differential (TD) across the insulated area, the economical thickness of insulation to use (N) can be determined.

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[5]Derivation of formula: $N=\sqrt{\frac{F_1 \times TD}{F_2}}$

$N = K/T \times 3.5^{-6} \times TD$

Where:

N=Thickness of insulation most economical at a given cost per board foot installed and a given TD

K=Thermal coefficient of the insulation

T=Thickness of insulation used

3.5^{-6}=Cost in dollars of absorbing 1 B. T. U. by refrigeration at $1.00 per ton

TD=Temperature differential across storage wall

Providing:

$K/T \times 3.5^{-6} \times TD = X$=cost of removing heat units (B. T. U.) passing into storage through each square foot of wall

and: The cost per board foot per hour for insulation is 1^{-6} dollars (based on an installed cost of 15.7 cents per board foot and a life of 18 years)

then: The cost of T inches is $T \times 1^{-6}$

Rewriting the equation:

$$\frac{K}{T \times 1^{-6}} \times 3.5^{-6} \times TD = X$$

Transposing and dividing by 1^{-6}:

$$\frac{K \times 3.5 \times TD}{T} = X$$

When:

X and T are equal, the cost of insulation just balances the cost of absorbing by refrigeration at $1.00 per ton the heat units passing through the insulation at the TD specified

Then:

$$\sqrt{K \times 3.5 \times TD} = X = T$$

Providing:

The insulation costs more or less than 1^{-6} per board foot per hour, the equation may be written: $\sqrt{\frac{K \times 3.5 \times TD}{F_2}}$

Where:

F_2=cost of insulation relative to the standard of 1^{-6} per board foot per hour, and the relations given above remain the same

Also where:

$F_1 = K \times$ cost of refrigeration as given in table

Then:

The equation can be written: $N=\sqrt{\frac{F_1 \times TD}{F_2}}$

[6]The thermal coefficient (K) denotes the number of British thermal units (B. T. U.) passing through 1 square foot of material 1 inch thick (1 board foot) in 1 hour per degree temperature differential (TD, or difference between the temperature on two sides of the material).

TABLE 3.—Table of factors used in formula $\sqrt{\dfrac{F_1 \times TD}{F_2}}$ for various thermal coefficients, insulation costs, and refrigeration costs

Thermal coefficient (K) per inch of wall and insulation cost per board foot installed	Factor 2	Cost of refrigeration per ton in dollars						
		½	¾	1	1¼	1½	1¾	2
		Factor 1						
0.02	0.13	0.03	0.05	0.08	0.09	0.10	0.12	0.14
.04	.27	.07	.10	.15	.17	.21	.24	.28
.06	.40	.10	.16	.23	.26	.31	.37	.41
.08	.53	.14	.21	.30	.34	.47	.49	.55
.10	.67	.17	.26	.35	.43	.52	.61	.69
.12	.80	.20	.31	.42	.52	.62	.73	.83
.14	.93	.24	.36	.49	.60	.73	.85	.97
.16	1.07	.27	.42	.56	.69	.83	.98	1.10
.18	1.20	.31	.47	.63	.77	.94	1.10	1.24
.20	1.33	.34	.52	.70	.86	1.04	1.22	1.38
.22	1.47	.37	.57	.77	.95	1.14	1.34	1.52
.24	1.60	.41	.62	.84	1.03	1.25	1.46	1.66
.26	1.73	.44	.68	.91	1.12	1.35	1.59	1.79
.28	1.87	.48	.73	.98	1.20	1.46	1.71	1.93
.30	2.00	.51	.78	1.05	1.29	1.56	1.83	2.07
.32	2.13	.54	.83	1.12	1.38	1.66	1.95	2.21
.34	2.27	.58	.88	1.19	1.46	1.77	2.07	2.35
.36	2.40	.61	.94	1.26	1.55	1.87	2.20	2.48
.38	2.53	.65	.99	1.33	1.63	1.98	2.32	2.62
.40	2.67	.68	1.04	1.40	1.72	2.08	2.44	2.76
.42	2.80	.71	1.09	1.47	1.81	2.18	2.56	2.90
.44	2.93	.75	1.14	1.54	1.89	2.29	2.68	3.04
.46	3.07	.78	1.20	1.61	1.98	2.39	2.81	3.17
.48	3.20	.82	1.25	1.68	2.06	2.50	2.93	3.31
.50	3.33	.85	1.30	1.75	2.15	2.60	3.05	3.45

An example will make clear the use of the table. If a storage wall is to have a thermal coefficient (K) of 0.2 B. T. U., or 0.1 less than the K factor for cork of 0.3;[7] the insulation is to cost 10 cents per board foot installed; the refrigeration is to cost $1.00 per ton; and the temperature differential (TD) will average 30° F.;[8] then factor 1 (F_1) would be 0.7 (column 5), corresponding to a thermal coefficient (K) for the wall of 0.2 (column 1) and a cost for refrigeration of $1.00 per ton (column 5); factor 2 (F_2) would be 0.67 (column 2), corresponding to a cost for insulation of 10 cents per board foot installed (column 1).

Entering these values in the formula as follows: $\sqrt{\dfrac{0.7 \times 30}{0.67}}$ and solving show that the economical thickness of the selected insulation would be approximately 5½ inches. The economical thickness of insulation would be increased by using a less expensive insulation, more costly refrigeration, or a higher average TD, and would be decreased by employing a more costly insulation, less costly refrigeration, or a lower average TD.

[7]The thermal coefficient (K) of a completed wall will always be less than the thermal coefficient of the insulation, because of the insulation value of that part of the wall, such as studs, sheeting, or masonry, not considered insulation. Therefore, the selection of the thermal coefficient (K) in table 3 should be 0.1 less than that of the insulation used.

[8]The temperature differential (TD) selected should be the average for the warmer portion of the storage season unless it is expected that the TD will be constant during the refrigerated period.

PROTECTING INSULATION FROM MOISTURE

Under certain conditions, the moisture mixed with the air may be condensed out as free water within the insulated wall.

The amount of water vapor which can be contained in a given volume of air depends upon temperature. The maximum amount of vapor which can be contained in any space increases as the temperature increases, and when a given space contains the maximum unit weight of vapor for a given temperature, it is said to be saturated. If it contains only one-half the amount that it is capable of containing, it is said to be 50 per cent saturated (50 per cent relative humidity). The ratio of the amount of vapor in any given space to the maximum amount that could be contained in that space is called the relative humidity. Since the maximum amount of vapor which can be carried in a given space is decreased as the temperature is decreased, it is obvious that if a space saturated with vapor is cooled, some of this vapor will be condensed in the form of free water. Also, if a space which is partially saturated with vapor is gradually cooled, it will in time reach its saturation temperature, and on further cooling, some of the water vapor will condense out into free water. If the final temperature is below the freezing point of water, ice will be formed.

It has been the practice for a good many years to apply some kind of good building paper on the sheathing or near the outside surface of the wall to protect the building from the outside moisture and weather. These papers are fairly good barriers against the transfer of air and water but not very efficient against the transfer of water vapor. At present, it is possible to purchase papers which are very efficient barriers against water vapor, but when they are placed on the wrong side of a wall, water vapor may increase in the wall as it enters from the unsealed side, and condensation may eventually occur. The addition of insulation in a wall increases the possibility of condensation within a wall, since this material offers little or no resistance to the travel of vapor but does reduce the temperature of the cold side of the wall and thus further increase the possibility of condensation in this cold portion.

A knowledge of the theory of vapor movement through walls is essential to the correct construction and protection of storage buildings and the insulation installed. Current research is uncovering many new facts in the proper construction of walls (4, 10, 15, 16, 26, 27). At present, authorities believe that moisture in the vapor form moves through a like mass of material which is permeable to vapor at a rate directly proportional to the vapor pressure gradient (the difference in vapor pressure between one side and the other) existing across the material in question and inversely proportional to the resistance offered by the material to the movement of vapor. In a storage wall, there are several materials, such as the wood used as sheathing and supporting members, which may take up moisture and be permeable or nonpermeable to vapor. Wood and other materials which are considered practically impermeable to water vapor contain a certain percentage of moisture, usually expressed on a dry-weight basis, which is directly related to the relative humidity of the air on each face and not to the absolute amount of water in the air, which is measured as vapor pressure. In this restricted case, moisture may move in the direction opposite to the vapor pressure gradient, providing the relative humidity is higher on the side of low vapor pressure. In storage walls, however, the primary concern is movement of moisture in the vapor form, and since vapor pressure rises sharply with rise in temperature, the moisture is almost always moving from the warm to the cold side of the wall. For this reason, it is customary to place the vapor seal on the warm side of the wall.

If no condensation of moisture took place in a storage wall, there would be no concern about which direction water vapor was moving. Since, however, the temperature in a wall varies with position in the wall, it frequently happens that somewhere in the wall, the temperature is at, or below, the dewpoint (temperature at which moisture condenses out of the air), and condensation occurs. That point of saturation is reached at a higher temperature with a higher relative humidity (table 9). For this reason, vapor barriers are used with walls to keep out the moisture in vapor form, thus lower the relative humidity of the air within the wall, and thereby lower the temperature to which the air may fall before condensation takes place. Since no perfect seal or vaporproof membrane is known, it is not practical to line both sides of a wall to eliminate moisture condensation within walls. The proper procedure is to install a good membrane on the warm side of the wall and no membrane, or one through which water vapor will pass very readily, on the other. Thus, what moisture does gain entrance to the wall will not increase the vapor pressure of the air in the wall to any appreciable extent, since it can escape more rapidly than it enters. Also, if condensation does occur in the wall, the free water will continue to evaporate and move on out of the wall toward the cold, dry air.

Unfortunately, placing insulation in a wall increases the tendency for condensation to take place in it, since the heat of the warm side does not penetrate nearly so far into the wall, and lower temperatures inside result. It has been found that many times more units of water will be condensed in an insulated and unsealed wall than in an uninsulated, unsealed wall at low temperatures. Sealing the wall on the warm side reduces this condensation, even in well-insulated walls, to a point where it can be ignored. Experience has also shown that no insulation, in itself, draws moisture into the wall and that if free water is kept from the insulation, what condensation does occur in the average wall will be at the boundaries between the insulation and the adjacent material. This condition is fortunate, for if vapor membranes unharmed by free water are used at the surface of the insulation, the insulation remains dry and effective and shows little tendency to deteriorate. Although the cost is seldom justified, an air space vented to the cold side on one or both sides of the insulation can be constructed to provide a condensing surface and a re-evaporating area in which the condensed moisture may continually evaporate and move to the outside under most favorable conditions.

Unfortunately for storage owners, especially those with refrigerated rooms, the direction of vapor movement is reversed between the early fall and midwinter. It is possible during early winter for the direction of vapor movement to reverse many times within as short a period as a day. This condition could occur only when the vapor pressures on both sides of the wall were about the same, however; so little vapor would be moving, and no harm could result. Harm might occur when the very warm side becomes the very cold side with a lowering of air temperature between September and January or February. The findings of those who have studied insulation in farm storages, both refrigerated and common, and have opened the walls occasionally during the storage season to examine the insulation, reveal, however, that in general, with materials properly installed, moisture does not accumulate to any damaging extent. The temperature differentials existing across storage walls are not great during most of the storage season and, consequently, the vapor pressure gradients are also slight, and moisture movement into and through walls is not rapid, nor is the dewpoint reached within the wall at all times. Therefore, only small quantities of moisture are condensed throughout the storage period, and with changing environmental conditions, much of this moisture is continually re-evaporated and passes on out of the wall, either toward the cooling coils or toward the cold, dry outdoor atmosphere. In addition, all apple storages stand idle for many months of the year when high outside temperatures drive out any trapped or absorbed moisture, and the insulation and supporting members therefore enter each succeeding season in good dry condition. Such conditions make the problem of adequately sealing storage walls much easier than it might otherwise be.

TABLE 4.—Vapor pressure in inches of mercury at six Ohio cities

Calculated from weather data*

	Cincinnati	Cleveland	Columbus	Dayton	Sandusky	Toledo
December	0.158	0.135	0.145	0.172	0.136	0.133
January	.154	.122	.140	.140	.120	.118
February	.134	.112	.129	.145	.113	.110
March	.206	.167	.189	.191	.155	.159
April	.260	.230	.255	.279	.236	.228
May	.392	.348	.376	.397	.350	.347
June	.517	.472	.505	.509	.489	.479
July	.586	.545	.583	.566	.502	.546
August	.574	.518	.551	.558	.539	.525
September	.472	.438	.448	.453	.431	.430
October	.314	.299	.308	.328	.256	.292
November	.212	.196	.202	.205	.192	.189
Mean	.332	.278	.319	.329	.295	.296
Years	20	30	25	10	15	29

*Climatological History of Ohio. 1924. W. H. Alexander. Bulletin No. 26, Engineering Experiment Station, The Ohio State University.

In tables 4 and 5 are given the mean vapor pressures calculated for various localities in Ohio and storages held at several temperatures and humidities. An inspection of these tables shows that during the fall and early winter, while the outside temperatures are higher than those in the storage, the vapor pressures are also higher outside than in the storage, and water vapor is moving into the storage. Later on, the outside temperatures and vapor pressures drop below the more or less constant values for the inside, the water vapor gradient reverses, and the vapor movement is toward the outside. The period during which the vapor is moving toward the inside of the storage is the greater. Therefore, it seems best, as has been recommended, to install a very good barrier on or near the outside of storage walls and to use a much more permeable seal or membrane on the inside of the wall. By this means, the movement of moisture into the wall from the outside during the relatively long period when the vapor pressure gradient is greatest is greatly reduced, and, at the same time, the movement of vapor through the inside of the wall later in the season is also somewhat restricted. What moisture does accumulate in the wall and insulation could eventually get out through the weaker seal. It is not so important in the common storage to use the seals as suggested for the refrigerated storage, since conditions are not so severe. In the common storage, the same good vapor seal can be used on both sides of the wall with satisfaction.

TABLE 5.—Vapor pressure, in inches of mercury, in storages with varying relative humidity

Storage temperature, degrees F.	Storage relative humidity, per cent					
	70	75	80	85	90	95
30	0.115	0.123	0.132	0.140	0.148	0.156
32	.126	.135	.144	.154	.163	.172
35	.143	.153	.163	.173	.183	.193
40	.173	.186	.198	.211	.223	.235
45	.210	.225	.240	.255	.270	.285

In table 6 is given the comparative resistance of various materials to vapor transmission. Certain paper-composition barriers are very effective vapor stops and are also inexpensive. The glossy surfaced asphalt-impregnated and surface-coated sheathing papers are excellent seals and sell for close

TABLE 6.—Relative rate of vapor movement through various materials*

Material	Relative rate of vapor movement
Data from Teesdale†	
Roll roofing, smooth surface, 40 to 65 pounds per roll, 108 square feet	1
Asphalt-impregnated and surface-coated sheathing paper, glossy surfaced:	
50 pounds per 500-square foot roll	3
35 pounds per 500-square foot roll	8
Duplex or laminated papers, 30-30-30	14
Duplex or laminated papers, 30-60-30	5
Duplex papers, reinforced	10
Duplex papers, coated with metal oxides	7
Insulation backup paper, treated	15
Plaster, wood lath	80
Plaster, gypsum lath	145
Plaster, three coats of lead and oil	27
Plaster, three coats of flat wall paint	31
Plaster, two coats of aluminum paint	8
Slater's felt	100
Plywood, ¼ inch, Douglas fir, soybean glue, plain	77
Plywood, two coats of asphalt paint	3
Plywood, two coats of aluminum paint	9
Insulating lath and sheathing, board type	215
Insulating sheathing, surface-coated	26
Compressed fiber board, 3/16 inch	36
Insulating cork blocks, 1 inch	44
Blanket insulation between coated papers, ½ and 1 inch	14
Mineral wool, 4 inches, unprotected	210
Data from Barre‡	
Asphalt-saturated felt, 15 pounds	4.2
Asphalt-saturated felt, 30 pounds	2.8
Sisalkraft, plain	1.8
Red rosin sheathing paper	191.5
White pine board, ¾ inch	4.4
Cedar bevel siding	5.7
Cedar bevel siding with three coats of white lead paint	3.2
Douglas fir plywood, ¼ inch	11.6
Douglas fir plywood, ¼ inch, with two coats of white lead paint	6.2
Commercial insulation board, ½ inch, sample 1	60.3
Commercial insulation board, ½ inch, with one coat of aluminum paint	38.2
Commercial insulation board, ½ inch, with two coats of aluminum paint	.0
Commercial insulation board, ½ inch, sample 2	34.2
Commercial insulation board, ½ inch, sample 3	90.4
Commercial insulation board, ½ inch, with one coat of aluminum paint, sample 3	21.7
Commercial insulation board, ½ inch, with two coats of aluminum paint, sample 3	1.1
Plaster on wood lath, three coats, ¾ inch	9.0
Plaster on wood lath, three coats, ¾ inch, with two coats of aluminum paint	.0
Brick wall laid up with mortar, 4 inches	2.5
Tile wall laid up with mortar, 4 inches	.4
Concrete, 1½ inches	5.3
Concrete, 1½ inches, with two coats of aluminum paint	1.6
Concrete, 1½ inches, with two coats of asphalt paint	5.0
Ground cornstalks, 2 inches	133.0
Rock wool, 1 inch	80.7
Sawdust, 2 inches	100.0

*Such data are variable. Method of testing and conditions surrounding material might alter figures given here. Data from Teesdale and Barre not comparable.

†(27).

‡(4).

to one-half cent per square foot. They are recommended for use on the outside of frame walls. Slater's felt transmits somewhat more vapor and is recommended for use on the inside of frame walls. These barriers should be applied vertically, with the edges lapping on the studs, after the insulation is installed and before sheathing. Horizontal joints should be made only where backed up with a plate or header. The barrier should be brought up tight against doors, ventilators, and other openings. On insulated ceilings, the barrier should be applied in a similar manner. If the ceiling insulation is left open to a ventilated space above, the insulation should remain dry. On masonry or tile walls, it is often easier to seal with paint. Ordinary paints of the flat wall, lead, or oil types do not offer as much resistance as frequently desired, but two coats of aluminum or asphalt paints seem satisfactory. Further details on construction and installation of insulation and vapor barriers will be found in Ohio Bulletin 573 (8). Present recommendations are, of course, made in the light of present knowledge and may be altered from time to time.

THE INSULATION OF FLOORS

The insulation of floors is essential for refrigerated storages. With the soil temperature between 50 and 60° F. and an attempt being made to maintain 30 to 35° F. in the storage room, there is considerable heat continually being dissipated by the floor and absorbed by the storage air and, in turn, by the cooling coils, when present. A discussion of the actual quantities of heat entering the storage through uninsulated and insulated earth floor is given later. The conventional method of reducing this heat load is to use the equivalent of 2 inches of cork between 3-inch layers of concrete on the floor. The floor construction is even more costly than that for walls and may be as high as 50 or 60 cents per square foot.

A new application of various cement-aggregate mixtures is suggested as an economical method of insulating storage floors and even walls. Occasionally in the past, an enterprising grower has mixed cement with sawdust and poured the mixture on floors and in walls of his storage. Recently, Skelton (24) has reported research with cement-sawdust concrete for use in poultry houses and dairy barns. The coefficient of thermal conductivity of a 3-to-1 mixture of sawdust and cement is reported as 0.60 to 0.70, which is close to twice that for cork; that is, this mixture is about one-half as efficient as corkboard, as tested under ideal conditions. There is no reason why wider ratios of cement and sawdust should not be used in floors to decrease the thermal coefficient of the mixture. A wider ratio than 3 to 1 will, of course, reduce the strength of the resulting slab and may make it unsuitable for walls. It appears that many other inexpensive materials, such as ground cork, ground corncobs, and blast furnace slag, can be used in similar mixtures. Further research is needed to establish the usefulnes of such mixtures in apple storages.

It is recommended that the area for floors be drained completely by means of tile and that a minimum of 6 inches of coarse material, such as cinders or stones, be applied as a base. Concrete may or may not be poured as a base for the insulating layer. If it is used, it should be sealed on top with an asphalt paint before the insulating mixture is poured. A thin wearing surface of smooth concrete directly on top of the new insulating mixtures is required, since they will scuff under severe service. Such construction keeps free water from entering the insulation but allows any moisture present during laying or condensed later, to vent through the wearing surface to the cold room air.

REFRIGERATION FOR THE FARM APPLE STORAGE

METHODS AND EQUIPMENT USED TO COOL APPLES

Mechanical refrigeration secured through the use of a compressor, refrigerant, and cooling and condensing coils or heat exchangers now provides the best and most economical method of cooling farm apple storages.

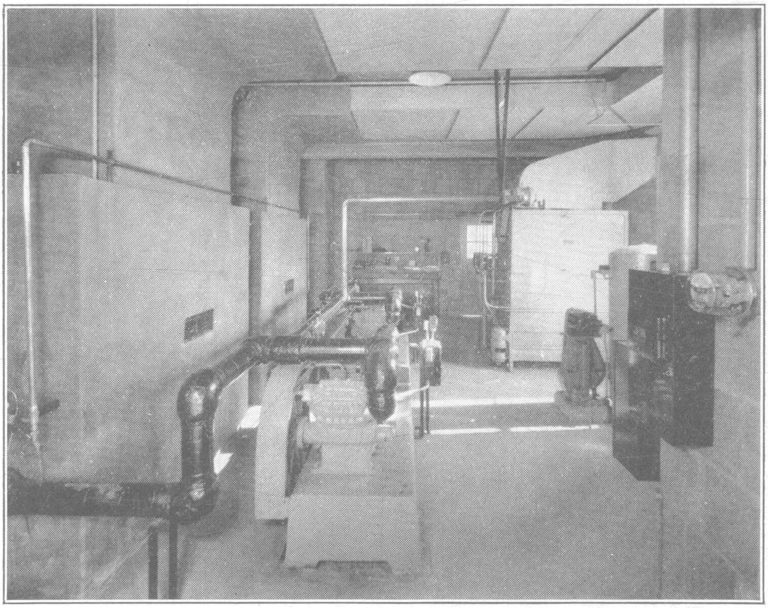

Fig. 2.—Complete refrigeration system for a 50,000-bushel apple storage

Two blower or forced-air evaporators on left; two 10 horsepower V-type four-cylinder compressors in middle; forced-air evaporative or shower-type condenser in right rear

There are many types of equipment and methods of cooling the storage air and delivering the absorbed heat to water or outdoor air. All systems have in common the compressor and two heat exchangers, one to absorb heat from the cooled product (evaporator), and one to deliver this heat to the outdoor air or water (condenser). A heat exchanger is any piece of equipment wherein heat flows from one medium to another medium at a lower temperature. The compressor acts as a vapor pump; that is, it sucks the refrigerant vapor in over its pistons, where the vapor is compressed and, thereby, raised to a higher temperature. The refrigerant is then delivered into the condenser, which, on being cooled by air or water, causes the compressed refrigerant vapor to condense. This cycle is completed when the compressed refrigerant liquid is allowed to expand in the cooling coils (commonly known as evaporators) in the storage room, where heat is removed from the storage air. A few of the larger systems circulate brine which, in turn, is cooled in a separate heat exchanger outside the storage room. Others spray the cooled brine over hanging cloth surfaces through which the storage air is forced by means of fans. In others, the brine is sprayed into a chamber in which cooling coils are suspended, and storage air is drawn over the coils and through the brine spray. Such equipment is more elaborate and costly than is justified for the moderate-sized farm storage. All that is needed in the average storage is a compressor, cooling coils, and a condenser. Various accessories discussed later are needed for the proper automatic functioning of the simplest system.

Figure 3 shows in detail the relation of the various parts of the equipment used today. The grower and storage operator should endeavor to understand the use and function of each part connected with a complete system, as such knowledge enables them to detect when the system is not functioning properly and to shut down the plant before costly damage or loss occurs. Usually, the operator will need to call in experienced help when repairs and finer adjustments need to be made.

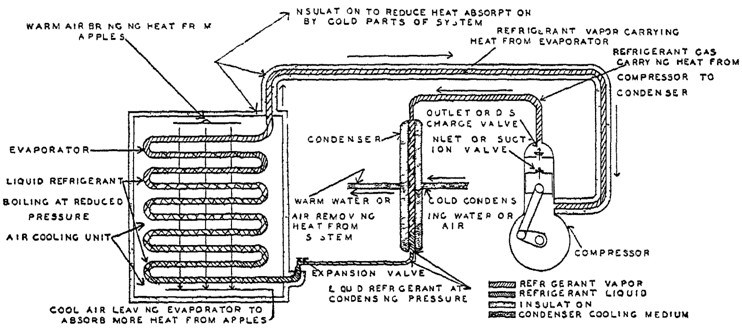

Fig. 3.—Elementary diagram of a typical compression refrigeration system

A classification of the various types of equipment might be as follows:

I. Direct-expansion
   A. Bare-pipe coils suspended from ceiling, walls, or center supporting posts. Storage air moves over the coils by gravity alone or aided with fans (fig. 10).
   B. Special coils with fins suspended from ceiling. Fans may be used.
   C. Specially compacted coils, usually with fins, installed in a cabinet equipped with blower fans (fig. 4). Such units may be ceiling or floor mounted. Several smaller finned coils with propeller-type fans attached may be suspended from the ceiling.
      1. Blower units may be equipped with brine spray nozzles to allow low temperatures to be maintained without frosting of the coils.
      2. Brine may be cooled, then sprayed into chamber through which storage air is drawn.

II. Indirect systems

Brine is cooled in a small tank unit with cooling coil; the cooled brine is then circulated through pipes in the storage room.

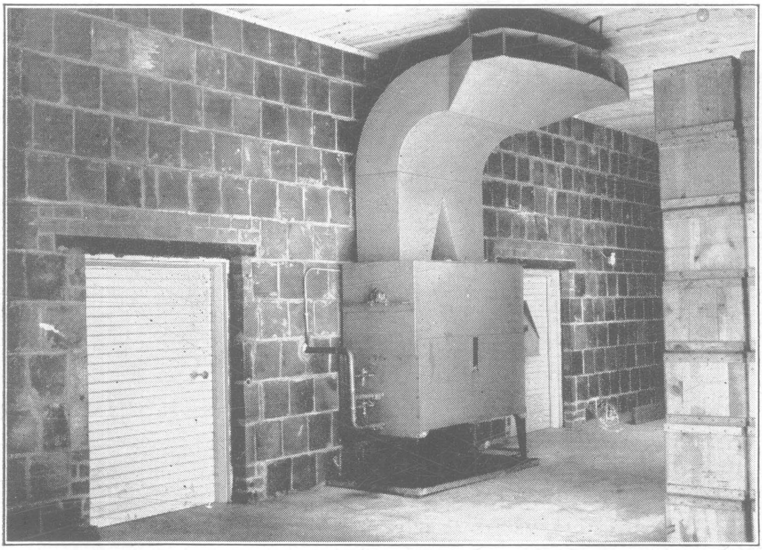

Fig. 4.—A cooling unit or evaporator installation in a 40- by 60-foot storage room

The fan motor is mounted on the right side and draws storage air up through the bottom of the cabinet, past the cooling coils (connected to the condensing unit in figure 7) and discharges it through the diffusion head at the ceiling. Note the water spray nozzle directly in front of the diffusion head and the smooth ceiling for obstructionless distribution of the cooled air to the far corners of the room without the use of ducts.

SELECTION OF EQUIPMENT

Considering the results secured in relation to first cost and economy of operation, only three systems are justified for farm storages:

The older, bare-pipe coil, direct-expansion system is still being installed in many storages. Its main advantages are: simplicity, freedom from leaks and frosting troubles, suitability for adjustment to any temperature, and long life. Its disadvantages are: poor control over humidity, slower cooling of product, possible freezing of product close to coils, no positive air circulation, possibility of air pockets and uneven temperature, and more room required.

There must of necessity be design compromise with any system, but the trend is definitely in the direction of the dry-coil, forced-air, floor-mounted unit, according to the refrigeration manufacturers who are installing equipment in Ohio storages. This equipment, properly applied, permits faster removal of heat from the fruit (20), supplies more uniform room temperature and moisture conditions, automatically maintains a higher relative humidity, lessens the growth of molds, aids in control of storage disorders, and seems to produce a better product when compared with gravity circulation evaporators (3). Approximately the same operating costs can be attained with proper selection and operation of equipment (11). The blower fans can be utilized to circulate outdoor air for cooling when desired.

Brine-spray cooling units possess only one advantage over the dry-coil types; that is, any temperature even below freezing can be maintained in the storage room with no frost collecting on the coils. This system permits continuous operation if desired and a raising or lowering of the room temperature at will. Contrary to popular belief, the fact that the room air is drawn through the wet brine spray has nothing to do with maintaining a high relative humidity, for the relative humidity depends solely upon the temperature of the spray and coil surface under any given storage condition. This phase of refrigeration is discussed elsewhere in this bulletin. Since very satisfactory installations are made with dry-coil units and since brine spray equipment and operating costs are higher, this equipment is not recommended except where cost is not considered. Salt brine is very corrosive, and some salt deposit on the fruit and room walls may occur. The life of metal materials in the room, especially air ducts, will be shortened. A new noncorrosive brine has recently been made available, however, and its use eliminates this objection to such equipment (11).

REFRIGERANTS

The four most commonly used refrigerants are listed in table 7, and their pressure-temperature characteristics given, as well as the relative size of compressor and relative horsepower required to deliver a unit of refrigeration.

TABLE 7.—Characteristics of the common refrigerants

Refrigerant	Gauge pressure in pounds		Relative size of compressors	Horsepower required to produce 1 ton of refrigeration
	At 5° F.	At 85° F.		
Ammonia	19.6	154.5	1.00	0.99
Freon (Dichlorodifluoromethane)	11.8	93.2	1.80	1 00
Methyl chloride	6 1	80.8	1.82	1.01
Sulfur dioxide	5 9*	51 7	2.45	1 01

*Inches of vacuum

Freon is fast replacing ammonia, which has been used more widely in the past than the other three, for use in apple storage cooling. Freon is an excellent refrigerant from many points of view, since it is odorless and harmless to humans and produce. It is more costly per pound than the others, however, and is difficult to keep in a refrigeration system. Its advantages far outweigh its disadvantages, though, and with some attention to testing for leaks occasionally, the final cost of Freon may be less than that of the other refrigerants.

Methyl chloride may be poisonous to humans in certain concentrations and is disagreeable in any concentration. It can be used interchangeably with Freon in compressors but is rapidly being replaced by the latter. Both refrigerants can be used with copper and copper alloy tubing and thus require less costly installations than ammonia, with which iron steel, or aluminum must be used.

Ammonia, although an efficient refrigerant, requires heavier equipment, and, with iron or steel piping, usually welded connections, to ensure against leakage of the refrigerant. Although small leaks of ammonia are quickly detected, the leaking vapor may reach sufficient concentration in the storage room air to injure apples (fig. 5).

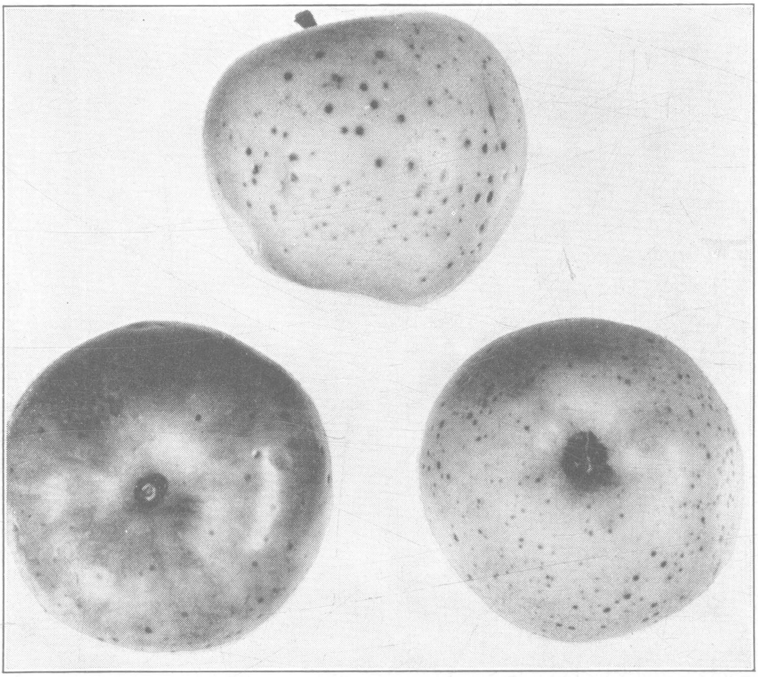

Fig. 5.—First stage of ammonia injury or burning on Baldwin

Ammonia vapor enters through lenticels (pores in the skin through which gases pass) killing the tissues directly surrounding the lenticels first.

Sulfur dioxide, like ammonia and methyl chloride, is disagreeable to use and is, therefore, being replaced by Freon.

Ammonia vapor requires the smallest volume per unit of refrigeration, and, although this property reduces the size of compressor needed for a given job, the more costly piping for blower unit and installation brings the cost close to that of a Freon compressor installation of comparable capacity. The horsepower required to produce a ton of refrigeration with ammonia is only slightly less than that required for Freon, and for all practical purposes, the four refrigerants may be assumed to be similar in this respect. Both ammonia and Freon equipment are giving satisfaction in their fields, and it is only a matter of opinion as to which will prove most satisfactory in any installation.

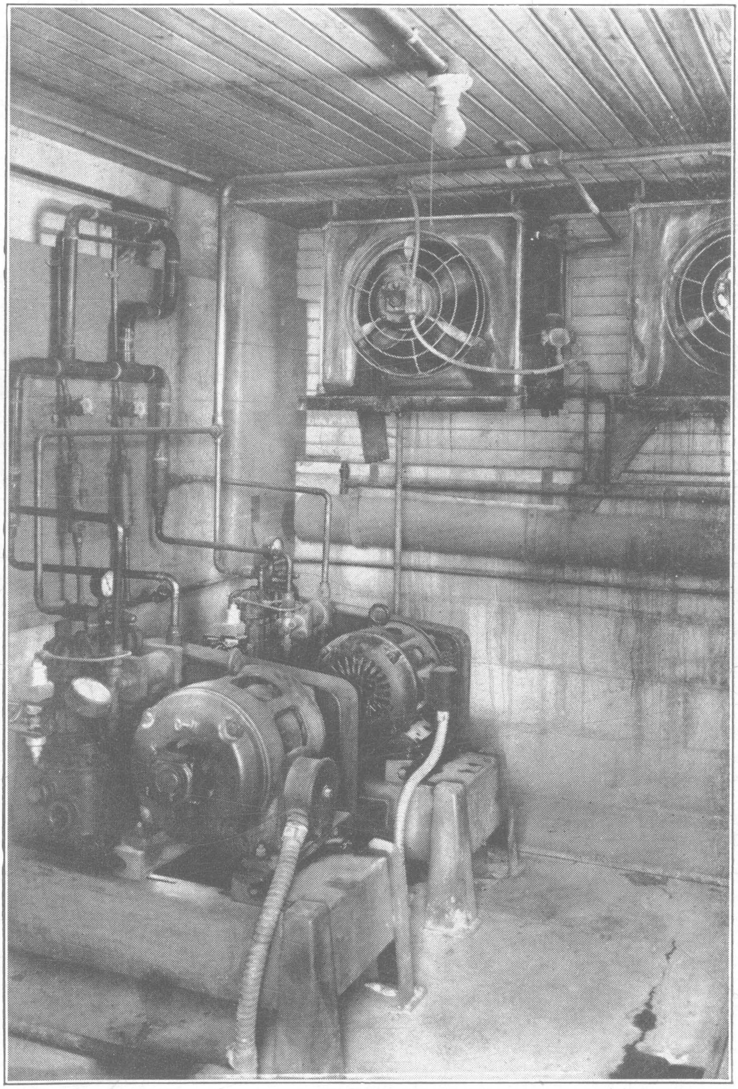

Fig. 6.—A compressor and condenser installation

Two cross-connected compressors with two-speed motors in foreground; two small evaporative-type condensers in windows, with long tank receiver below, in top background. This installation allows maximum flexibility in operation, a complete stand-by unit during winter operation, and air-cooling of condensers without water during freezing weather.

AIR DISTRIBUTION IN STORAGE

Ducts for air distribution are needed only in the large storage rooms or where several rooms are cooled with one floor-mounted unit. The centrifugal or blower-type fans used in all such units are capable of delivering air under reasonable pressure for considerable distances. Air is frequently blown 40 feet or more where no obstruction exists and will usually be delivered to the farthest wall in most storages, especially when the unit is placed in the center of the room. A diffusing head must be used to direct the air to all parts of the room in equal proportion (fig. 4). Where ducts seem advisable, they may be constructed of wood or wood and galvanized metal. A smooth, unobstructed ceiling is required where no ducts are used and, therefore, supporting beams must run in the direction of air flow, or the ceiling boards must be brought flush with such beams. Occasionally it is less expensive to construct a duct than to alter a ceiling for proper air distribution.

CONDENSING EQUIPMENT

There may be some confusion as to what equipment is included in the condensing portion of a refrigeration system, because the word "condenser" is frequently applied to a refrigeration compressor, as well as to the condenser proper, which may be incorporated in the compressor frame or be an entirely separate unit mounted some distance from the compressor. The condenser receives the compressed refrigerant vapor (from the compressor) at a relatively high temperature and, by means of water or air passing over it, the hot vapor is condensed to liquid with no reduction in pressure. All the refrigerant

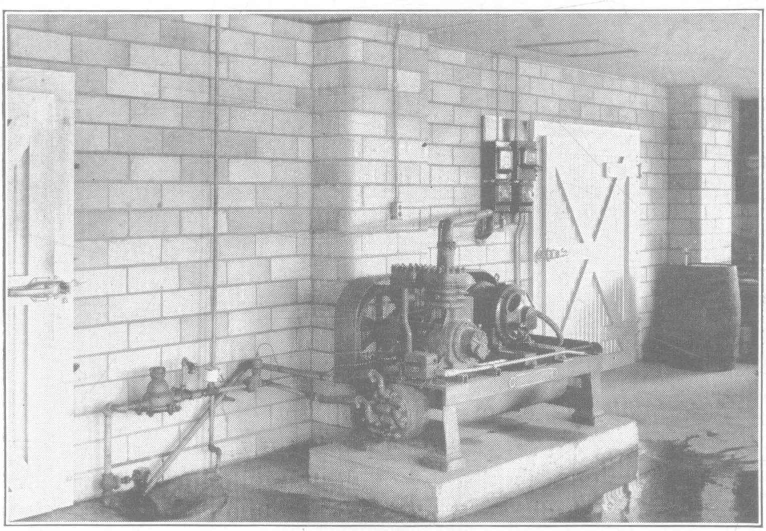

Fig. 7.—A refrigeration unit installation in an apple storage salesroom

The cooling unit or evaporator is shown in figure 4. Note the shell-and-tube condenser cooled by a city water supply in the base of the compressor mounting.

not in the evaporator, compressor body, or lines accumulates in the condenser or connected tank, called the receiver. The condenser and receiver can be considered one unit, although they may be some distance apart (fig. 6). It is from the receiver that the liquid refrigerant passes in the liquid line to the expansion valves at the intake of the evaporators. It is the expansion (or boiling) of the liquid refrigerant in the evaporator which absorbs heat and does the actual cooling (fig. 3, 4).

Small refrigerating units (up to 3-ton capacity) can be purchased with water- or air-cooled condensers. Larger units have water-cooled condensers or use some water with air in what is known as the evaporative condenser. It is becoming the practice with units using the newer refrigerants, such as Freon, to mount the compressor, condenser, and controls together as a single unit. There are some advantages in having the complete unit assembled at the factory, where precautions can be followed in removing from the castings and system all moisture, which may cause considerable damage in connection with some refrigerants. Such compact units require less room and are ready to connect to the evaporator for cooling the produce (fig. 7).

COMPRESSORS

There are many types of compressors on the market. They differ in size and weight per ton of capacity with the refrigerant employed. Ammonia compressors are usually operated at slower speeds, are heavier than other types, and are most frequently two-cylinder models (fig. 8).

The compressors using the newer refrigerants are smaller and lighter and operate at higher speeds with less vibration (fig. 6, 7). They are more frequently multiple-cylinder machines and may use the V-type cylinders. A great deal of flexibility is obtainable with such light equipment, for it is common practice to mount two small compressors on one frame and belt them to one motor which may operate at two speeds. By cutting out one compressor or two cylinders of a four-cylinder machine and by altering the speed of operation, refrigeration capacity can be altered automatically to suit the varying load. Two or more small compressors cost little more than one large one, and for this reason are frequently interconnected to one evaporator so that the size of the compressor can be reduced when the cooling load drops during the cold winter months. Compressors are usually designed to operate at any one of three different speeds or revolutions per minute.

There are many minor variations in such parts as valves, refrigerant seals, bearings, and the like. A reliable company will guarantee and repair its equipment. Company reputation is of more importance than the specific characteristics of the equipment, since minor changes in design are being made continually, and what is good today may be improved tomorrow.

Compressors are rated on a basis of horsepower rather than tons of refrigeration, since their capacities will vary with the temperature of the refrigerant, the cooling medium surrounding the condensing coils, and the speed of rotation. For example, a compressor using Freon refrigerant and operating at constant speed and condensing temperature will deliver 9.75 tons of refrigeration at a refrigerant gas or vapor temperature of 10° F., and 20.8 tons at a gas temperature of 30° F. The same machine at twice the speed will produce practically twice as much refrigeration. The average Freon compressor usually has more than twice as much capacity at a suction or refrigerant temperature of 53° F. as at 9° F. Thus it is obvious that compressor capacity increases with suction temperature and corresponding back pressure (table 8). The advantage of using forced-air evaporators is thus evident, for with such equipment, satisfactory room temperatures can be maintained with higher suction or refrigerant temperatures in the coils, and correspondingly higher

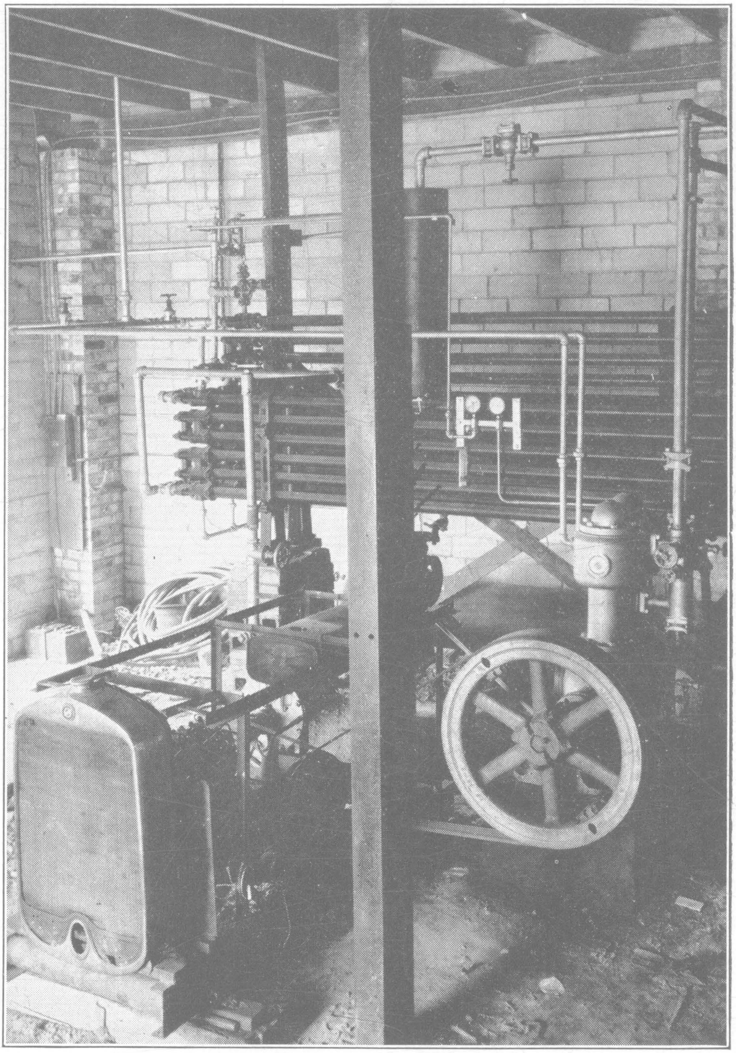

Fig. 8.—An ammonia compressor, V-belt driven by an old automobile engine

The engine also powers a small centrifugal water pump (below the gasoline tank) used to circulate water through the condenser pipes (in background) and spray nozzles on the roof above (see fig. 9).

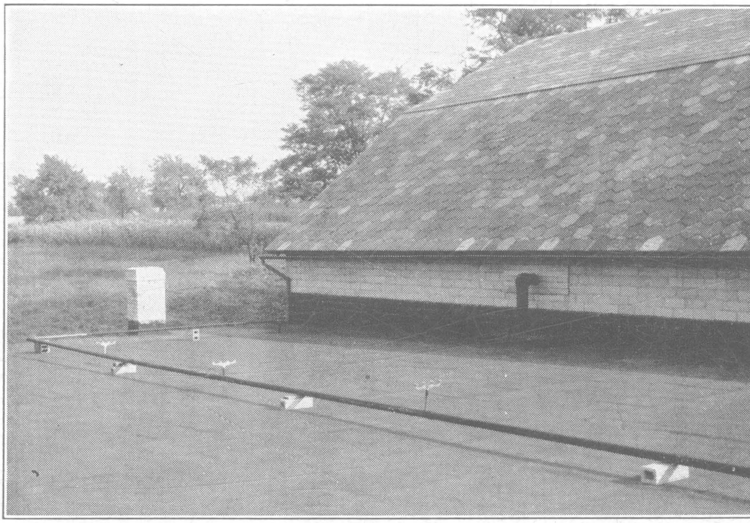

Fig. 9.—A simple and inexpensive condensing water-cooling system for the ammonia compressor and condenser shown in figure 8

Water issuing under pressure from the old orchard spray nozzles is air-cooled and drains from the roof to a storage tank below.

capacities are secured from the compressor used. Such a system may be contrasted with bare-pipe coil ammonia systems, in which slow gravity movement of air over the cooling coils requires that much lower refrigerant gas temperatures be maintained in the coils to cause the air to move with some appreciable speed.

TABLE 8.—Increase in capacity of a refrigeration unit with rise in room temperature

Refrigerant (Freon) temperature in degrees F.	Corresponding suction or back pressure in pounds per square inch, gauge	Relative capacity
0	9	1.00
20	21	1.63
32	30	2.10
42	39	2.57

CONDENSERS

The purpose of the condenser is to remove heat from the hot gas as it leaves the compressor cylinders and thereby condense it into liquid form. On compression, all gases are heated to temperatures corresponding to the pressure applied. When the hot gas is cooled slightly, it will condense into a liquid, in which form it flows into the receiver in readiness to be re-expanded in the evaporator. The heat removed at the condenser originates in the storage room and is absorbed by the refrigerant vapor in the evaporator. The heat produced in compressing the gas or vapor from the evaporator suction pressure and temperature to the condenser pressure and temperature is removed at the condenser.

The type of condenser depends upon the method of keeping the condensing surfaces cool. Water-cooled condensers are compact drums, frequently concealed beneath the body of the compressor by the frame members (fig. 7) or supported on the wall (fig. 8). They may be termed "shell and tube", "shell and coil", or "shell and finned tube." The cooling water usually circulates within the tubes while the refrigerant is discharged into the drum, where it condenses on the water-cooled coils and fins. The cooling water consumption varies with conditions but for purposes of rapid estimate may be considered 70 to 90 gallons per hour per ton of refrigeration. Air-cooled condensers consist of finned-tube radiators into which the hot gas is discharged and in which the cooling air is blown over the fins by means of fans. The hot gas gives up its heat through the metal to the cooler air, condenses on the inside surfaces, and drains to the bottom and a connected receiver. Small units are cooled with forced air alone from propeller-type fans. Some units are equipped with a water line and nozzle arrangement whereby atomized water is blown over the coils with the air to aid cooling (fig. 6). Considerable heat is removed from the coils and contained refrigerant to evaporate the water, as well as to warm the air, and greater efficiency is secured. In the larger units, the condenser coil is mounted in a cabinet with a blower or forced-air fans and several spray nozzles. The units employing water to aid cooling are termed "evaporative" or "shower" condensers. The similarity between such units and forced-air cooling units is apparent.

It is very important that the circulating air of the "evaporative"-type condenser be discharged in such a manner (to the outdoors) that it cannot be recirculated back into the condenser, since the discharge air is practically saturated with moisture and would not take up additional moisture upon its recirculation.

Although more power is required with an evaporative condenser, to move water, as well as air, the water requirements are much less than those of the shell-and-tube type of condenser. Only approximately 5 per cent as much water is consumed, and where water is costly, the saving is considerable. In addition, as the outdoor air temperatures lower in the fall and winter, the quantity of water used is reduced until no water is required, and the danger of freezing water lines is eliminated. Where the water supply is limited or costly or may freeze in the winter, evaporative condensers are one answer to the problem.

CONDENSING WATER SUPPLY

An ample supply of clean water must be available for the peak requirements of condenser operation. Water can be secured from municipal supplies, shallow or deep wells, streams, rivers, or ponds. Water pumps of sufficient capacity must be provided in connection with all condensers not connected to a municipal line. Water treatment and scale (deposit left by the water on the condenser pipes) removal methods are available for maintaining the efficiency of the equipment.

It is desirable that the water temperature be as low as possible, for then less water is required, and lower head or condensing pressures result. This reduction in condensing pressure reduces the power required to operate the compressor and condenser and increases the capacity of the system for a given power input.

The automatic water valve admitting water to the condenser on demand requires a pressure of 20 to 25 pounds per square inch gauge for operation. If lower pressures only are available, an electrically operated solenoid valve can be used.

COOLING TOWERS

Cooling towers to cool condensing water for recirculation purposes are sometimes mounted on the storage building roof and are dependent upon the evaporation of a small amount of water to cool the remaining water in the tower to a point a few degrees below that maintained in the condenser. Considerable water must be circulated through the condenser with this method, since the towers are designed to cool water only through a range of approximately 10° F. It is possible to cool the water to within a few degrees of the wet-bulb temperature of the air (table 9).[9] When the air circulating through the tower is near saturation, very little cooling is accomplished (table 9). Occasionally, forced-draft towers are used, but when power is required for forced-air fans, the evaporative condenser is usually employed.

COOLING EQUIPMENT (EVAPORATORS)

The portion of any refrigeration system which actually absorbs the heat from the storage room air is known as the cooling unit (fig. 4, 10). The heat is absorbed by the cold refrigerant within the unit and transferred to the condensing unit to be dissipated into the cooling medium used with the unit. Cooling units used in storages are termed "evaporators", since they are containers in which the liquid refrigerant "evaporates" or boils. The evaporator coils act as heat exchangers between the storage air and the refrigerant. The real cooling is accomplished as the liquid refrigerant evaporates or boils and in the process absorbs a large quantity of heat. This heat is the latent heat of evaporation and is similar to the heat absorbed by water as it evaporates, or boils, at 212° F. The main difference is that water will not boil at room temperature and that heat must be artificially supplied to it, whereas all refrigerants boil at temperatures considerably below those maintained in apple storage rooms at normal or atmospheric pressures or below. The words "direct expansion" are frequently applied to refrigeration systems in which the refrigerant "expands" or "evaporates" directly in the cooling coils placed in the storage room in contrast to those systems in which the refrigerant expands in coils immersed in brine, and this brine is then circulated through pipes or coils in the storage room.

[9]The wet-bulb temperature of the air is the temperature indicated by a thermometer, the bulb of which is surrounded by a wet wick, and is, therefore, cooler than that of the dry-bulb thermometer because of the heat absorbed from the thermometer bulb in evaporating the water in the wet wick. The difference between the wet- and dry-bulb temperature depends upon the rate of evaporation, which, in turn, depends upon the relative humidity of the air surrounding the wet wick At saturation, or 100 per cent relative humidity of the air, the wet- and dry-bulb temperatures are the same.

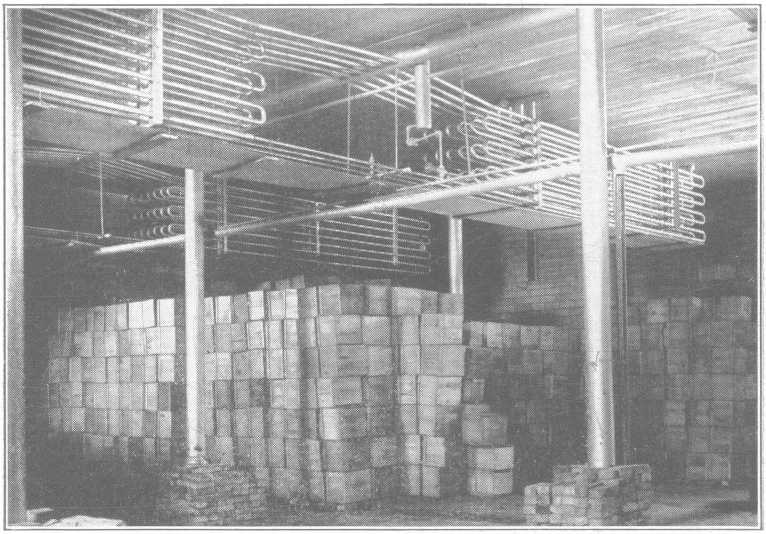

Fig. 10.—Direct-expansion ammonia cooling coils connected to compressor in figure 8.

Note that each vertical tier of coils is directly connected to the large pipe suction manifold (top center) and liquid line manifold (directly below). All joints are welded to eliminate possibility of ammonia leaks. Drip pans are sheets of corrugated galvanized metal supported on wood cleats wired to the coils. A common house gutter and downspout carry drip water to the storage drain. This system of cooling coil installation provides the maximum efficiency by maintaining a full-flooded coil at all times.* Heat exchange between the storage air and cooling coils is increased by propeller fans supported on hinges in the openings in the walls (right rear), and they can be used to cool with outdoor air when desired.

*A full-flooded coil is one in which some liquid refrigerant is present at all times. Since it is the boiling of the liquid refrigerant which absorbs heat, each full-flooded coil is absorbing heat throughout its entire length when the refrigerant vapor is being removed by the compressor.

In order that cooling may take place, the refrigerant and "evaporator" coils must be maintained at a lower temperature than the storage air; that is, heat transfer depends upon the basic fact that heat always flows from a warmer to a colder object.

The amount of heat that can be absorbed by a cooling unit depends upon three factors: the rate of heat movement through the evaporator walls, the area of the cooling (evaporator) surface, and the temperature differential (temperature difference between the inside and outside of the evaporator, or the T. D. of the evaporator). The product of these three factors gives the heat transferred in a unit of time, which is usually expressed as British thermal units per hour. Since the heat transferred is the simple product of these three factors, changing the value of any one of them will change the amount of heat transferred.

As with any given evaporator the area of the cooling surface and its rate of heat transfer are fixed, the quantity of heat absorbed depends upon the remaining variable factor, temperature differential (T. D.). For this reason, the basic rating of all evaporators is given in B. T. U. absorbed per hour per degree of temperature differential. Any coil will absorb twice as much heat at a T. D. of 20° as at one of 10°; that is, the heat absorbed will vary directly as the T. D. varies. These fundamental relations can be expressed in an equation as follows: $H = K \times A \times T.D.$, where H represents B. T. U. per hour; K is the coefficient of heat transfer; A is the total cooling unit area; and T. D. is the temperature difference between the refrigerant and the air being cooled.

TABLE 9.—Relation between relative humidity and temperature difference between room, dew point, and refrigerant with a room temperature of 35° F.

Relative humidity	Room temperature			Maximum allowable cooling effect of coil	Probable temperature difference between room and refrigerant
	Dry-bulb	Wet-bulb	Dew point or saturation		
Per cent	*Degrees F.*	*Degrees F.*	*Degrees F.*	*Degrees F.*	*Degrees F.*
95	35	34.5	33.5	1.5	5.1
91	35	34.0	32.5	2.5	6.4
86	35	33.5	30.5	4.5	7.9
82	35	33.0	30.0	5.0	9.6
77	35	32.5	29.0	6.0	11.5
73	35	32.0	28.0	7.0	13 6
68	35	31.5	26.5	8.5	15.9
64	35	31.0	25.0	10.0	18.4

The temperature differential for apple-cooling equipment is of the greatest importance, for it controls the moisture conditions that prevail in the storage room. A large temperature differential means a greater cooling of the air as it passes over the evaporator, and when the dewpoint temperature of the air is reached, some of the moisture it carries will condense out on the cooling coils. The higher the relative humidity of the air and the lower its temperature, the more readily will the dewpoint or condensing temperature be reached. In apple storage rooms, where the object is to carry high relative humidities and low temperatures, the tendency of the cooling coil to condense moisture is considerable. Some of the condensed moisture re-evaporates at the coils and is carried back into the storage by forced or gravity air movement. The majority of the condensed moisture, however, drains into the drip pans beneath the cooling coils or is held as frost on the coils. If this moisture is allowed to drain out on the storage floor, it should eventually return to the air, and such a procedure is highly recommended. Too many storage operators who do not realize this or who dislike wet floors will drain all drip water to the outside. This water comes from the apples, as well as into the room from outside, and if it is not replaced by water piped into the storage, its loss may cause noticeable shriveling of the fruit.

In tables 9 and 10 are given data to show the relations existing between relative humidity, air temperature, and cooling coil temperature under selected storage conditions. In table 9 are given the dewpoint or saturation air temperatures existing in a storage room held at 35° F. and several levels of relative humidity. Both the wet-bulb and dewpoint temperatures lower with lowering relative humidities at a constant room temperature. For example, at 95 per cent relative humidity, the room temperature or the temperature of any object, such as storage wall, ceiling, floor, crate, or fruit, cannot drop more than 1.5° F. without having moisture condense out of the air as fog or free moisture. Since most solid bodies will be slightly cooler than the surrounding air, which is moving, condensation takes place on these bodies before the dewpoint of the moving air is reached, and this condensation tends to prevent fog from forming in the storage. Air at 86 per cent relative humidity can be lowered from 35 to 30.5°, or 4.5° F., before condensation will occur; if no moisture is to be removed from the air passing over the cooling coil, however, that coil must not be more than 4.5° cooler than the air. In blower or forced-air units, the air is moving over the coils and fins rather rapidly and, therefore, does not reach the temperature of the coil. With such units, the temperature of the cooling coil can be several degrees below the dewpoint temperature of the air without lowering the fast-moving air to that temperature. In the last column of table 9 is given the probable temperature difference between the room air and the refrigerant which can occur before condensation will take place. This is termed the temperature differential of the coil and, along with many other minor factors, governs its cooling, as well as its dehydrating, effect.

TABLE 10.—Moisture removed from 1 cubic foot of dry air with a temperature of 35° F. upon cooling*

Temperature drop or cooling effect of coil	Moisture content of delivered air at 85 per cent relative humidity	Relative humidity of return air required to maintain delivered air at 85 per cent relative humidity	Moisture removed from 1 cubic foot of 35° F. air with a relative humidity of—		
			70 per cent	80 per cent	90 per cent
Degrees F.	*Grains per cubic foot*	*Per cent*	*Grains per cubic foot*	*Grains*	*Grains*
1.5	1.91	80	0	0	0
2.5	1.83	76	0	0	.01
4.5	1.66	69	0	0	.21
5.0	1.62	67	0	.02	.26
6.0	1.58	65	0	.02	.26
7.0	1.53	63	0	.12	.36
8.5	1.40	58	.03	.27	.51
10.0	1.36	56	.08	.32	.56
20.0	.65	27	.91	1.15	1.39

*"Dry" air refers to air containing no free moisture; it may, however, contain moisture in the vapor form to the point of saturation.

In table 10 are given data to show the grains of moisture removed from 1 cubic foot of 35° F. air entering the cooling coil at three different relative humidities and cooled 1.5 to 20° F. below the entering air temperature. In addition, the relative humidity of the air entering the coil which must be maintained to cause the delivered air to be at 85 per cent of relative humidity through the same temperature range is included. In the second column of table 10 is given the reduction in the actual or total amount of moisture in the air leaving the cooling coil as its temperature is lowered. This reduction occurs because low-temperature air at 85 per cent relative humidity contains less total moisture than air with the same relative humidity at a higher temperature. However, as long as the relative humidity remains constant, a lowering of the air temperature does not increase water loss from the fruit; in fact it may decrease such loss by lowering the vapor pressure deficit (difference in vapor pressure) between the internal atmosphere of the fruit and the storage air. In the third column of table 10 is given the relative humidity of the air entering the cooling coil unit at 35° F. required to maintain an 85 per cent relative humidity of the air leaving that unit. Air at the higher humidities upon cooling will leave the cooling unit at relative humidities above 85 per cent, and air entering at lower humidities than shown in column 3 will leave the unit with less than 85 per cent of moisture, on a relative basis. In columns 4, 5, and 6 are given the grains of moisture per cubic foot of air removed from the entering air at 35° F. and 70, 80, and 90 per cent relative humidity. When no moisture is removed, the air may leave the cooling unit with more moisture than when it entered, providing some free moisture is present on the coil. In the remaining cases, moisture is condensed from the air onto the coils and drips to the floor of the unit. The quantity condensed depends upon the relative humidity of the entering air and the cooling effect of the coil, and increases with increasing relative humidity and temperature differential across the cooling coil. Table 10 shows that if 35° F. storage air with 70 per cent or more relative humidity is cooled 8.5° F. or more, some moisture will be condensed out on the cooling coil. Storage air is usually closer to saturation than 70 per cent, and most cooling units cool storage air more than 8° F., certainly more than 5° F. For this reason, frost collects on cooling coils rather readily, and it places an added load on the compressor to absorb the latent heat of condensation.

The problem of maintaining the desirable high relative humidity in storage air is difficult. It can be approached two ways: by using considerable coil surface operated at a low temperature differential, or by spraying moisture into the storage air. The former method increases the initial cost of the installation, and the second places an additional load on the cooling equipment which is not reflected in cooler fruit. Obviously, a compromise is necessary, and it should be made through the purchase of as large a cooling surface as is required to maintain approximately 85 per cent relative humidity in the storage room during the holding period. The data in tables 9 and 10 show that a very satisfactory relative humidity is maintained when the air is cooled 4 or 5° in its passage over the coil. This amount of cooling would require a temperature differential between the refrigerant and air temperature of from 8 to 10° F. for blower equipment. Where gravity air circulation is used, it is not feasible to employ sufficient coil surface to maintain a temperature differential of 8 to 10° F., and it is better to add moisture to the storage air by artificial means, since frost on coils used with gravity air circulation does less harm than on coils used with forced-air fans. Sixteen to 20° F. might be chosen as a temperature differential for gravity air circulation equipment. Wider temperature differentials than these will provide good refrigeration but somewhat less control over humidity. It is a matter of design standard, and each fruitgrower must decide which standard of performance he will pay for and see to it that he gets this performance before he accepts the refrigeration plant.

Table 11 shows the proportion of the total refrigeration which may be ascribed to absorbing the heat given up by the water vapor in the air as it condenses to free water on the cooling coils. Condensing 1 pound of water releases approximately 1,000 B. T. U. of heat, and the refrigeration required to absorb it adds nothing to the cooling of the apples and is pure waste unless the water condensed is made to evaporate again in the storage room. When this happens, heat is absorbed from the air and fruit, and the process becomes merely one of heat traveling from fruit to air to water vapor and thence to the cooling coils. For this reason, it is recommended that the drip water from the cooling coils be run out upon the storage floor for re-evaporation or pumped through pipes to spray nozzles. The same objective is secured by using water from any other source. However, applying heat from some artificial source to vaporize the water would be defeating the purpose of employing heat generated within the storage for the vaporization.

TABLE 11.—Proportion of total refrigeration load due to latent heat of condensation in cooling 35° F. air at three relative humidities

Temperature drop or cooling effect of coil	Per cent of total refrigeration load consumed in condensing moisture from 35° F. air at—		
	70 per cent relative humidity	80 per cent relative humidity	90 per cent relative humidity
Degrees F.			
1.0	0.0	0.0	0.0
2.0	.0	.0	.0
3.0	.0	.0	.0
5.0	.0	.0	37.3
7.0	.0	19.5	37.7
9.0	.0	25.3	37.2
11.0	15.4	27.6	36.6
13.0	19.3	28.6	35.9
21.0	23.5	28.5	33.0

From the data in tables 9, 10, and 11, it can be concluded that a compromise on the level of relative humidity maintained is necessary for economical operation of the storage and that 85 per cent or slightly above has proved satisfactory for the keeping of fruit. At present, therefore, refrigeration equipment is designed to maintain this humidity automatically, at least during the holding period. Future experiments may prove that a higher relative humidity is desirable.

AUTOMATIC HUMIDITY CONTROL WITH PNEUMATIC WATER NOZZLES

Experience with humidity control through the use of spray nozzles and water under pressure show this method to be undesirable because of the large quantity of sprayed water which settles on the fruit, containers, and floor. Others (3), have shown that if air under pressure is supplied to special nozzles connected to a water supply under atmospheric pressure, a very fine atomization of the water occurs and none settles out of the air to wet the surroundings. Such equipment is easily constructed out of any small air compressor and a container with an automatic water level control, such as the float valves used on various types of equipment for supplying drinking water to livestock. Worn-out compressors from domestic refrigerators make excellent air compressors. Air and water piped separately around the storage room allow the distribution of the atomizing nozzles throughout the storage. A constant supply of water under atmospheric pressure is maintained in the lower of the two pipes by means of a tank and float at the same level at one end of the storage room or adjacent to it.[10]

[10]Binks Manufacturing Company, 3114 Carroll Ave., Chicago, Ill. Monarch Mfg. Works, Inc., Salmon and Westmoreland Sts., Philadelphia, Pa.

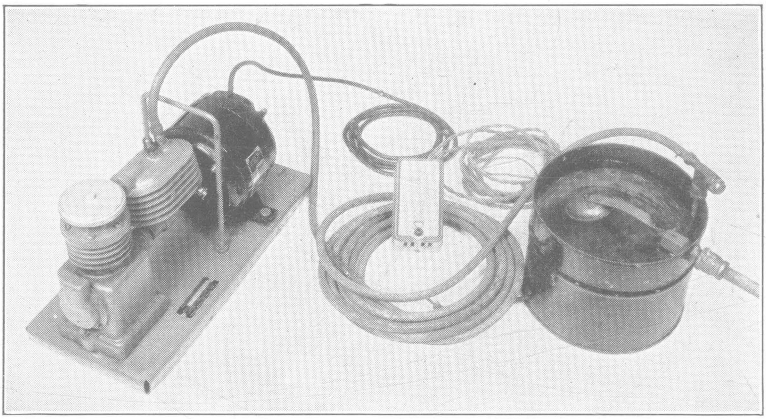

Fig. 11.—Complete apparatus for humidifying an apple storage

Small air pump (left), humidistat control (middle), water tank with constant level float valve and pneumatic water spray nozzle (right)

OPERATION OF COOLING EQUIPMENT

FORCED-AIR COOLING UNITS

When properly applied, forced-air cooling units are preferred for apple storages. As already pointed out, such equipment can be operated with narrow temperature differentials, which will result in higher average moisture conditions, greater capacity from condensing unit, and faster and more uniform cooling of fruit. In new storages, lower ceilings can be used with this type of equipment, and where the head room in existing storages is limited, such units are used to advantage, also. More capacity is available in rooms in which this type of equipment is used. The simplicity of installation and the elimination of coil, decks, and baffles make the cost of forced-air equipment favorable to its use.

The proper application is more exacting with this type of equipment, however, primarily because for efficient operation, only the minimum of frosting may occur. A coil choked with frost will not allow air to pass through, and no cooling of the storage air will occur under such conditions. The first requisite is that the cooling coil or evaporator be selected to balance the hourly capacity of the condensing unit, and not the hourly refrigeration load. In turn, the condensing unit or compressor capacity is chosen to equal or exceed the hourly refrigeration demand.

There are several interrelated factors, such as length of storage period, proper temperature, moisture, and air distribution, and proper capacities and operating temperatures, which are of importance in refrigeration. These factors must be properly balanced against each other if a successful installation is to be made. Temperature differential between storage air and refrigerant has already been discussed. The effect of various air velocities on the dehydration of fruit is unknown at present. It is known, however, that air with a high relative humidity will cause less shriveling, even at high velocities, than air with a low relative humidity. It is also known that apples can be subjected to higher air velocities and a wider temperature differential with less harm when they have just entered storage than during the holding period, when the fruit is cooled to 32 or 35° F. This is an important fact, since it is best to extract the field heat from the product rapidly, and higher air velocities, especially in tightly packed rooms, hasten this heat extraction. It is important at all times to install forced-air cooling units so that no direct air blasts from the units strike the fruit. The cooling is accomplished by the air moving through the stacked crates on its return to the unit. Smooth ceilings are required, and fruit cannot be stacked above the levels of the forced-air outlet, 1½ to 2 feet below the ceiling, or it will be subjected to a blast of air from the blowers. Ducts are usually not essential, and they add to the installation cost.

Occasionally, manufacturers employ variable-speed fans, or fan economizers, which turn off the fans entirely for part of the day. These are difficult to use, though, for as the fan speed is varied, the capacity of the cooling unit is varied, and the change may upset the necessary balance between condenser, evaporator, and, in turn, the temperature differential and storage room conditions. It is generally recommended that the blower fans be operated only at one speed and continuously. Fewer controls are required, and the defrosting portion of the cycle is completed. From the standpoint of the cooling equipment, there is no reason why the fans should not be stopped during that period of the off cycle when there is no frost on the coil. However, it is imperative that the fans operate for an indeterminate period of time after the compressor has stopped in order that any frost present will be quickly melted from the coils. It is also a decided advantage when cooling apples to re-evaporate as much of the clinging moisture on the coils as possible rather than have it drop to the drip pan beneath the unit and be carried away in the drain. More favorable humidities may be expected when blower fans operate continuously. For these reasons, fan economizers are not widely used with apple storage equipment.

FROSTING AND DEFROSTING OF FORCED-AIR COOLING UNITS

Probably the most important consideration in the use of forced-air equipment for apple cooling is that of frost formation and removal, for holding temperatures for apples are very close to the minimum permissible for automatic defrosting of dry-coil forced-air equipment. These coils are suitable for maintaining much lower temperatures when manual defrosting is practiced, or if various automatic controls are employed as accessories to ensure complete defrosting under all loads and outdoor temperatures. If the design or sales engineer has a correct understanding of the problem, no difficulties should be encountered. However, a tendency to overlook the problem or underestimate its importance does occur, either because of the competitive angle in equipment sales or because the salesman overestimates the knowledge of the fruitgrower concerning refrigeration equipment. Most storage operators believe that the equipment they are purchasing will do anything they ask of it, and expect such performance on an entirely automatic basis.

The best installations are those in which capacity and coil surface are both sufficiently large to handle any cooling load which may be placed upon them without an undue lowering of the refrigerant temperature or widening of the temperature differential between the coil and storage room air temperatures. A condensing unit (compressor and condenser combined) large enough to carry a given cooling load might be connected to a small cooling coil, and, by employing a wide temperature differential between the coil and room air, the cooling load could be handled. However, the wide temperature differential would tend to cause a more rapid and greater accumulation of frost on the coils, and frequent cycling would be necessary to defrost the coil. The efficiency of the complete system would thereby be reduced, for besides the lost refrigeration due to frequent off periods, the frost accumulating on the coil would act as an insulator and reduce the efficiency of the coil. Frost also restricts the free passage of air over the coil. Condensing the excess moisture on the coil adds nothing to fruit cooling unless that moisture is re-evaporated into the storage air. A larger coil would, of course, alleviate the trouble, but at a greater initial cost. To aid in reducing the deleterious effects of frost, coils with relatively wide fin spacing (three fins per inch) should be used. It is the balance between the compressor and evaporator (cooling coil) that is of greatest importance. Providing the compressor is not oversized in comparison with the evaporator, a particularly heavy heat load will merely raise the refrigerant temperature and place a greater load on the compressor motor. No frost will form on the coils, however. Undersizing the compressor in relation to the cooling coil surface will never cause harm, but the equipment may not then be capable of handling the required cooling load.

There are two distinct possibilities in selecting equipment for apple storage houses where the forced-air dry-coil type is recommended. With the proper design, an air temperature of 35° F. can be maintained, and any frost collecting will automatically disappear on the normal off-cycle period of operation.

It is advisable to design the equipment for an operation period of not more than 18 hours out of 24. During the pulldown period there are possibilities that the equipment may be required to operate more than 18 hours in 24 to absorb the heat from the storage being loaded rapidly. During this period, manual defrosting may be resorted to, or an automatic control can be installed. If temperatures below 35° F. are desired during the holding period, some method of positive defrosting must be employed. In normal installations, frost will readily form on evaporator surfaces in air below 32° F., and the rate of formation will depend upon the moisture in the air (table 10). It is a usual practice to install somewhat more evaporator capacity to offset the increased accumulation of frost between defrosting periods for installations below 35° F.

There are several methods of ensuring a defrosted coil:

The system can be designed with the proper balance between evaporator and condenser capacity and storage room load and utilize evaporator coils large enough to maintain a narrow temperature differential between coil and room air (8 to 10° F. during holding period and up to 30° F. during pulldown period). Such a system will operate at a 35° F. room temperature and defrost on the normal off cycle of the compressor. This method of defrosting is desirable where it can be used, as it is accomplished automatically by the use of controls normally supplied with most condensing units. Occasionally the equipment might have to be defrosted manually if the fruit were placed in storage too rapidly. Where temperatures lower than 35° F. are desired, or where undercapacitated equipment is to be used, one of the following methods of defrosting must be employed.

The evaporator can be defrosted when necessary by manual means. If it is permissible for the room temperature to rise to above 35° F., merely pulling the compressor switch will cause the coil to defrost in due time. If it is essential that the room temperature should not rise appreciably, it is necessary to install short ducts from the intake and discharge openings of the cooling cabinet to the warmer air of an adjacent room or to the outdoors. By the turning of dampers, the warm air can be made to circulate over the coil and defrost it rapidly. This method is seldom used, since the slight rise in storage air temperature essential for defrosting increases the daily mean apple flesh temperature only slightly.

A time clock can be used to turn the compressor off and on at predetermined intervals and eliminate any manual attention to defrosting.

The best method of ensuring complete defrosting is to use auxiliary controls. All condensing units used in apple storage cooling are equipped at the factory with controls which stop the compressor whenever the pressures on the two sides of the unit (evaporator and condenser) get either above or below the setting of the controls. These controls are provided primarily for maintaining the proper limits of efficient operation and as safety measures. When used with a room thermostat, they afford automatic defrosting. However, if the load is suddenly increased for any reason, or if a change in outdoor conditions takes place, frosting may occur unless the controls are readjusted, especially where the evaporator coil is too small for peak loads or there are other adverse conditions, such as unusually high air humidities or lack of refrigerant in the system. In order to overcome any conditions tending to aggravate frosting and to ensure that all frost is off the coils before the cooling portion of the refrigeration cycle is resumed, two types of controls have been devised. In one, a low-pressure cutout switch has been so combined, by means of a special electrical circuit, with a room thermostat that even though the thermostat circuit is closed and the room is needing refrigeration, the compressor cannot start until the evaporator coil reaches a predetermined pressure (and temperature) which is sufficiently high to ensure complete defrosting of the coil. The thermostat then controls the room temperature by stopping the compressor when the low temperature at which it is set is reached, and starting the compressor when the temperature of the room rises to the starting setting, providing the pressure switch contacts are closed. The second control operates entirely on the temperature of the coil and room air. Instead of the pressure of the coil actuating a switch, the temperature of the coil does it. Thus, the coil must warm to a predetermined temperature (where frost will melt) before the switch contacts are closed. Since for every pressure of refrigerant in a cooling coil there is a corresponding temperature, the two controls are very similar in their method of securing a defrosted coil and as low a room temperature as possible with the equipment to which the control is attached. Both controls are supposed to cause cycling of the compressor under all conditions, but the stopping of the compressor when a given amount of frost has collected is not as accurately controlled as the starting of the compressor. Usually any low-pressure control normally attached to the condensing unit will stop the compressor when frost accumulation on the coil causes a drop in pressure and temperature to the cutout setting of the control. Some improvement in the control of the compressor just as the frost forms, would be desirable.

Defrosting with a spray of water or brine is feasible but not advised when simple controls to do the job otherwise are available. Where temperatures of 32° F. or lower are maintained, it is undesirable to allow the room temperature to rise to 34 or 36° F. to ensure defrosting, and spray or outside air types of defrosting must be used.

It is apparent that with proper equipment, properly installed, automatic defrosting can be secured with finned coils maintaining temperatures at or above 34 or 35° F. with no special equipment. The coil surface must be at 32° F. or below for frost to collect on the coil, and the coil temperature must be raised to the same point or above to remove the frost accumulation by melting. Lower average room temperatures are more difficult to obtain, although with outside air or spray defrosting, they are maintained with dry-finned coils. It is poor economy to select small coils because of their low initial cost. The larger coils allow higher refrigerant temperatures to be maintained while handling a given cooling load, and since the higher the refrigerant temperature, the greater the capacity of the condensing unit, power is conserved through a shortening of the operating time. Occasionally selecting a large coil makes possible the use of a lower-priced condensing unit than could be used if a smaller coil were purchased. In addition, higher humidities and easier, more rapid defrosting are secured by using coils with large surface areas.

RUNNING TIME OF FORCED-AIR COOLING UNITS

For automatic defrosting it is essential that the compressor and coils have sufficient capacity to cool the room and its contents with a maximum of 18 hours of total running time per day. For apple storage work the load is at a peak for a relatively short period during the time loading is taking place. Therefore, it seems desirable to size the equipment on the basis of 24-hour, or continuous, operation for peak load periods and resort to manual defrosting when necessary. In this way, the size of the equipment can be kept at a minimum, and its operation will be entirely automatic for practically the entire storage season.

Also, economy can be secured in any system by the flexibility possible through altering the operating speed of condensing units. By changing pulley sizes on the compressor motor, the capacity of the unit can be stepped up for more rapid pulldown in the early fall, and it can be reduced by the same means for the holding period.

Control of the suction pressure, that is, the refrigerant temperature, also offers the possibility of obtaining greater or less capacity from the same units. The use of two-speed motors or automatic unloading devices on compressors has already been suggested.

A typical example of the operation of a small plant and the simple calculations involved in understanding its operation might be as follows. A condensing unit operating on an 18° F. suction or refrigerant temperature might have a capacity of 3 tons of refrigeration. Since a ton of refrigeration is equivalent to the melting of 1 ton of ice at 32° F. to water at 32° F., it can be expressed in B. T. U. The heat absorbed in melting 1 pound of ice at 32° F. (or the heat released when 1 pound of water freezes to ice at 32° F.) to water at that temperature is 144 B. T. U. For 2,000 pounds, or 1 ton, it is 288,000 B. T. U. The time in which this melting or freezing takes place is 24 hours per ton of ice. Thus, a 1-ton refrigeration machine would absorb 288,000 B. T. U. per day, 12,000 B. T. U. per hour, or 200 B. T. U. per minute. The 3-ton machine used as an example could absorb 36,000 B. T. U. per hour or 600 B. T. U. per minute. The cooling coil or evaporator to which it is connected also has a heat-absorbing rating. This rating is given as a basic rating, or the heat it will absorb per hour per degree of temperature differential between the air passing over the coil and the refrigerant temperature. In this example, the coil has a basic rating of 2,000 B. T. U. Therefore, if the compressor can absorb 36,000 B. T. U. per hour, the coil would require an 18° F. temperature differential ($\frac{36,000}{2,000} = 18$) just to balance the compressor; that is, the air being drawn over the coil would need to be 18° F. warmer than the refrigerant in the coil for that coil to absorb 36,000 B. T. U. per hour. The heat supplied to the cooling coil comes from the air, which, in turn, absorbs heat from the room and its contents, such as apples, crates, and walls. In this example, the air of the storage is passing over the coil at the rate of 5,500 cubic feet per minute, and it has a temperature of 36° F. and a relative humidity of 85 per cent. Since the cooling of 55 cubic feet of air 1° F. releases 1 B. T. U., the cooling of 5,500 cubic feet of air 1° F. would release 100 B. T. U. per minute, or 6,000 B. T. U. per hour. With the evaporator coil and condenser capable of absorbing 36,000 B. T. U. per hour, the 36° F. air entering the cooling unit would be cooled 6° F. ($\frac{36,000}{6,000} = 6$) and thus be discharged at 30° F. Naturally, in actual practice there might be some slight variation due to such additional heat sources as the heat of condensation of moisture out of the air, the heat coming from the electric motor driving the blower fans, and the like.

Several changes might be made in the system. If the speed of the blower fan were increased and more air per minute drawn over the coils, the air would not be cooled quite 6° F., and the discharge temperature of that air would rise. Slowing up the speed of the air would lower its discharge temperature. However, the final cooling effect would be the same, providing the return air remained at 36° F. and the coil temperature at 18° F. The increased speed of movement of the storage air would give less time for that air to absorb heat from the apples and other objects in the storage, and the temperature of the air returning to the unit would not rise as rapidly as it would when the air was moving less rapidly. The final result is always a product of the amount of cooling in degrees Fahrenheit by the rate of that cooling in cubic feet of air per minute. This fact explains why the temperature in bare-pipe coil evaporators with gravity air movement is held so much lower than that in the evaporators of forced-air units. With the former, the air moves so much more slowly across the coils that it must be cooled to a lower temperature to produce the same results in a given time as would be secured with the forced-air system. If the apples are not supplying 36,000 B. T. U. per hour to the storage air, the return air temperature will drop below 36° F. and soon reach a low point at which the room thermostat may stop the compressor motor. If the heat load should stay constant or even rise, the compressor would run continuously unless stopped manually. In the fall, when apples may be loaded more rapidly than the refrigeration system is designed to handle, room temperatures do rise somewhat, but they are reduced again during the next lull in loading. Such a heavy load increases the capacity of the refrigeration system, since the suction pressure and refrigerant temperature rise, and a more or less automatic compensation for the increased load occurs. At the same time, a greater load is placed on the compressor motor, and for this reason it is important that an overload release be provided for all refrigeration systems.

As the storage season progresses, the load on the cooling system is greatly reduced. It is then that it is economical to reduce the speed of the compressor by use of a two-speed motor or by changing the pulley size. When two compressors are employed, one may be stopped entirely. To prevent too rapid lowering of refrigerant temperature and its resulting short cycling of the compressor and aggravation of the frost problem, some system of balancing out the system toward the lighter loads must be followed. The best results will be secured when the system is kept close to a balance and too long or too short periods of operation prevented. If the cooling coils do frost, the proper automatic controls will ensure their periodic defrosting and thus tend to compensate for poor balance of the storage heat load and hourly capacity of the cooling system.

In the sample refrigeration system under discussion, there is sure to be frost accumulation with an air temperature of 36° F., an 85 per cent relative humidity, and the coil at 18° F. The return air contains 2.1 grains of moisture per cubic foot, and since the discharge air at 30° F. can contain only 1.9 grains per cubic foot when completely saturated, 0.2 grain of moisture per cubic foot, or over 9.4 pounds of water ($\frac{5{,}500 \times 0.2 \times 60}{7{,}000} = 9.4$) per hour will be left on the coils or in the drip pan, if the return air is maintained at 85 per cent relative humidity. Providing this occurs during the loading period, it is not so serious. However, during the holding period, such a condition indicates too wide a temperature differential (18° F.) between return air and coil temperature, and a temperature differential of from 8 to 10° F. would be much more satisfactory. With no gain or loss of moisture in the air of the storage, the continual condensing of moisture at the coil would in time lower the storage air to 75 per cent of relative humidity. Moisture is probably continually evaporating into the air from the wet coils and drip pan, as well as coming from the apples, and possibly through the walls, ceiling, and floor. Moisture could be added by hose or spray nozzles, but it is best that a narrow temperature differential be used so that little or no moisture will actually condense from the air.

With the assumed system, the air must be cooled 6° F. to absorb the 3-ton refrigeration load. Enlarging the coil or increasing the capacity of the condensing unit would not alter the necessity of cooling 5,500 cubic feet of air per minute 6° F. to extract the necessary 3-ton heat load. It is the number of degrees that the return air temperature is lowered that affects the moisture removed from that air. Some gain in efficiency would be obtained by the increase in coil area and increase in condenser capacity, but the moisture relations are more important with apples. To remove the 3-ton heat load without lowering the air 6° F. but only 4° F., the blower fans would need to be increased in speed to deliver 8,250 cubic feet per minute instead of 5,500 cubic feet per minute. If the air then moved too fast in storage, it would be necessary to install two forced-air cooling units to handle the load at reduced temperature differentials. Since the original 18° temperature differential is not too great for apples during the loading period, the size of this unit would not be increased, nor would the fan speed be increased. After the loading of the storage was completed and the cooling load dropped rapidly below the 3-ton level, it would be wise to reduce the pulley size on the compressor motor if a two-speed motor were not a part of the equipment. By this means, the capacity of the cooling system could be reduced to more nearly the actual cooling load. If the load were only one-half of the former value, or 18,000 B. T. U.

per hour (1.5 tons), the suction or refrigerant temperature could be raised to 27° F. and the air cooled only 3° F. A decided improvement would be secured in the condenser capacity, running time, and moisture conditions in the room. The return air at 36° F., being cooled only 3° F., would leave the coil at 33° F. and over 95 per cent relative humidity, and no moisture would be condensed out on the coils. The temperature differential between the return air (36° F.) and the refrigerant (27° F.) would be 9° F., or just between the 8 to 10° F. specified for good design. Frosting should be no problem.

THE HEAT LOAD ON AN APPLE STORAGE REFRIGERATION SYSTEM

The purpose of a refrigeration system in an apple storage is to absorb the heat in the storage. A state of equilibrium is established between the sources of heat and the capacity of the refrigeration unit to absorb and dissipate that heat. There are two phases to consider in refrigerating a storage. The first is the process of lowering the temperature of the storage room and its contents to a desired point. This is termed the "pulldown" heat load. The second phase is holding that selected temperature, which tends to rise because of the heat being conducted from places of higher temperature and from physiological sources, such as apple respiration, and mechanical sources, such as electric lights and motors. This second phase is termed the "holding" heat load. The pulldown heat load is always greater than the holding heat load, and the pulldown heat load includes the sources of heat making up the holding load. A list of pulldown and holding heat load sources might include:

Sources of heat during—

Pulldown period

I. Dead load from—
   A. Heat extracted in lowering the temperature of—
      1. Storage walls, ceiling, and floor heated by contact with outdoor air and earth
      2. Contents of storage
         a. Apples
         b. Apple containers
         c. Partially saturated air
         d. Free water
         e. All remaining contents of storage

II. Heat leak load from—
   A. Infiltration
   B. Opening doors
   C. Lights
   D. Electric motors on evaporators
   E. Heat leak inward through walls, ceiling, and floor
   F. Heat of condensation of water

III. Live load from—
   A. Heat of respiration of fruit
   B. Human occupants

Holding period

I. Heat leak load

II. Live load

It is customary to speak of refrigeration loads in terms of heat leak load, fan load, dead load, and live load. To the total of these loads may be added 10 per cent for the remaining small heat loads not included or not determined, and for unusual conditions.

In calculating the various heat loads, certain temperatures must be chosen. The design temperature refers to the temperature maintained in the storage room and may be different during the pulldown and holding periods. The ambient temperature refers to the temperature outside the storage walls, ceiling, and floor and naturally varies with these three locations and with the time of year. The temperature of the incoming fruit is also important. It may be lower for fruit entering storage early in the morning than for that coming in later in the day.

CALCULATING THE APPLE STORAGE HEAT LOAD

The total peak heat load is calculated separately for the pulldown and holding periods. The peak, or greatest daily, load is of first importance, since the refrigeration system which is capable of handling the pulldown peak daily load will be more than adequate for the lesser holding load. The heat leak, dead, and live loads together make up the peak pulldown load.

HEAT LEAK LOAD CALCULATIONS

To calculate the heat leak, it is necessary to use a heat leak factor which represents the quantity of heat in B. T. U. transmitted through a square foot of barrier per day per degree of temperature differential across the barrier. When the thermal coefficient for various materials, including air and surface resistances, is known, it is a simple matter to calculate such heat leak factors for any type of construction. Since it is customary to think of all storage walls as made up of structural material and added insulation, it is convenient to calculate heat leak factors for representative walls, floors, and ceilings without insulation and for the same walls with various thicknesses of the commoner types of insulation. Surprisingly enough, the heat leak factors are very similar for all walls of equal thickness, regardless of type of construction and insulation used. Exceptions are walls using some loose-fill insulations, such as ground cork, sawdust, shavings, and other less costly materials. In table 12 are given heat leak factors for walls of various thicknesses insulated with several common materials.

Since thermal coefficients for walls, ceiling, and floor are frequently given in B. T. U. per hour instead of per day and can be calculated for any of these surfaces, the data in table 13 are presented to give approximately the refrigeration load per 1,000 square feet for any storage. The heat leak factors in table 12 divided by 24 can be used to enter table 13.

The product of the heat leak factor, insulated area, and temperature differential gives the daily heat leak in B. T. U. For example, with a storage wall 60×10 feet and an outside air temperature of 85° F., if the insulation supplied gave a heat leak factor of 2, the calculation would be as follows: $60 \times 10 \times 2 \times (85° - 35°) = 60{,}000$ B. T. U. per day. This calculation can be made for other walls, ceilings, and floors by using the appropriate figures. The sum total of such calculations for any storage constitutes the total heat leak in B. T. U. per day and can be converted to tons of refrigeration by dividing by 288,000 B. T. U., the heat absorbed by a refrigerating unit with a capacity of 1 ton.

TABLE 12.—Heat leak factors for storage walls, ceiling, and floor with various thicknesses of the common insulation materials*

Material	Thickness of insulation, inches	Heat leak through wall, ceiling, or floor as B.T.U. per 24 hours per square foot per 1° F. difference in temperature between the two exposed surfaces
Corkboard and other board products, shredded redwood bark, spun glass, rock wool, other proprietary loose-fill materials	1	4.7
	2	3.5
	3	2.7
	4	2.3
	5	2.0
	6	1.8
	7	1.6
	8	1.5
Ground cork and cement-cork mixtures of 1 to 6 ratio and above†	1	6.0
	2	4.5
	3	3.5
	4	3.0
	5	2.5
	6	2 3
	7	2 0
	8	1.8
Sawdust, shavings, and other farm by-products	1	13.0
	2	9.5
	3	7.5
	4	6.0
	5	5.5
	6	5.0
	7	4.5
	8	4.0

*Insulation well protected from moisture penetration as free water or vapor.
†Tentative; reliable test data not yet available.

TABLE 13.—Approximate refrigeration load in tons per 1,000 square feet for various thermal coefficients and temperature differentials

Thermal coefficient	Temperature differential											
	5	10	15	20	25	30	35	40	45	50	55	60
	Tons of refrigeration											
0.02	0.01	0.02	0.03	0.04	0.05	0.06	0.07	0.08	0.09	0.10	0.11	0.12
.04	.02	.04	.06	.08	.10	.12	.14	.16	.18	.20	.22	.24
.06	.03	.06	.09	.12	.15	.18	.21	.24	.27	.30	.33	.36
.08	.04	.08	.12	.16	.20	.24	.28	.32	.36	.40	.44	.48
.10	.05	.10	.15	.20	.25	.30	.35	.40	.45	.50	.55	.60
.12	.06	.12	.18	.24	.30	.36	.42	.48	.54	.60	.66	.72
.14	.07	.14	.21	.28	.35	.42	.49	.56	.63	.70	.77	.84
.16	.08	.16	.24	.32	.40	.48	.56	.64	.72	.80	.88	.96
.18	.09	.18	.27	.36	.45	.54	.63	.72	.81	.90	.99	1.08
.20	.10	.20	.30	.40	.50	.60	.70	.80	.90	1.00	1.10	1.20
.30	.15	.30	.45	.60	.75	.90	1.05	1.20	1.35	1.50	1.65	1.80
.40	.20	.40	.60	.80	1.00	1.20	1.40	1.60	1.80	2.00	2.20	2.40
.50	.25	.50	.75	1.00	1.25	1.50	1.75	2.00	2.25	2.50	2.75	3.00
.60	.30	.60	.90	1.20	1.50	1.80	2.10	2.40	2.70	3.00	3.30	3.60
.70	.35	.70	1.05	1.40	1.75	2.10	2.45	2.80	3.15	3.50	3.85	4.20
.80	.40	.80	1.20	1.60	2.00	2.40	2.80	3.20	3.60	4.00	4.40	4.80
.90	.45	.90	1.35	1.80	2.25	2.70	3.15	3.60	4.05	4.50	4.95	5.40
1.00	.50	1.00	1.50	2.00	2.50	3.00	3.50	4.00	4.50	5.00	5.50	6.00

It is unwise and poor economy to refrigerate a storage with a poorly insulated floor. At least, the equivalent of 2 inches of corkboard should be provided in the floor. An uninsulated floor may produce 48 B. T. U. per square foot per degree of temperature differential per day. With an earth temperature of 50° F. and the storage air carried at 35° F., there would be 720 B. T. U.

per square foot of floor per day, or close to 2.5 tons of refrigeration capacity for a 5,000-bushel storage (table 2) requiring from 3.1 to 4.8 tons of refrigeration (table 14). Four inches of concrete would reduce the floor heat load by one-half. An uninsulated or poorly insulated floor may account for one-fourth to one-half the total peak refrigeration load, and almost the total load during the winter portion of the holding period. Those who wish to calculate the floor heat load under any assumed condition can use a thermal coefficient of 2 for an earth floor and 1 for a floor covered with 4 inches of concrete. Many operators cool with outdoor air rather than pay the power bills incurred during the colder months, when a large heat load from the uninsulated floor is almost the sole load on the refrigeration system.

DEAD HEAT LOAD CALCULATIONS

The dead load on a refrigerating unit represents the heat stored in the apples, containers, walls, and the like. It is always wise to cool the storage walls prior to loading the storage by operating the refrigeration system at least 3 to 7 days while the room is empty. Even though the air temperature may drop rapidly to the design storage room temperature, heat will continue to move from the surroundings into the storage air, and unless some time elapses, the walls will not be cooled sufficiently, and storage room air temperatures will rise again rapidly when the refrigeration is shut down. After the walls are cooled, only heat leak from the outside must be absorbed to maintain the design room temperature. For this reason, dead load calculations include only the heat contained in the incoming fruit. This heat is termed "sensible", since as the heat is removed from the fruit, one can "sense" its lowered temperature. The calculations of sensible heat load include the weight of the fruit and containers, the number of degrees through which the fruit is lowered in temperature (temperature differential), and the specific heat of the fruit. Specific heat represents the amount of heat which must be absorbed per pound of material to reduce its temperature 1° F. The standard is water with a specific heat of 1. Apples are largely water (75—85 per cent), and a figure of 0.9 is used as the specific heat of this fruit. An example of the calculation of dead load is as follows: For an assumed loading rate of 200 bushels per day and the crate and apples weighing 50 pounds, the calculation is: $200 \times 50 \times (80°—35°) \times 0.9 = 395{,}000$ B. T. U. per day. The peak dead load, therefore, is dependent upon the number of apples loaded into storage per day, the temperature of those apples, and the extent to which they are cooled in 1 day. This dead load peak is not always reached the first day, and it may be delayed for several days.

In figure 12 are given the daily refrigeration loads for two assumed installations. The smaller of the two is based on 3,000 bushels of apples held at 35° F. with a loading rate of 200 bushels, or 6⅔ per cent, per day. The larger storage was considered to have a capacity of 85,000 bushels held at 32° F. with a loading rate of 6,000 bushels, or 7 per cent, per day. Both graphs show the rapid increase in refrigeration load to nearly a peak in 4 days and a continual but slower rise up to the fourteenth or fifteenth day, when the storage is filled. The loads then drop off very rapidly to only a fraction of their peak levels. In the very small storages only does the heat leak load exceed the dead load, which usually constitutes the greatest single load on the refrigeration unit. Therefore, the rate at which any storage is loaded is the primary factor in selecting the size of equipment required to refrigerate that storage. It is expedient to load a storage as slowly as is consistent with picking operations, and since some days no fruit enters storage, the refrigeration system may then cool a greater number of bushels of apples on those days when the picking and loading rate is somewhat above normal.

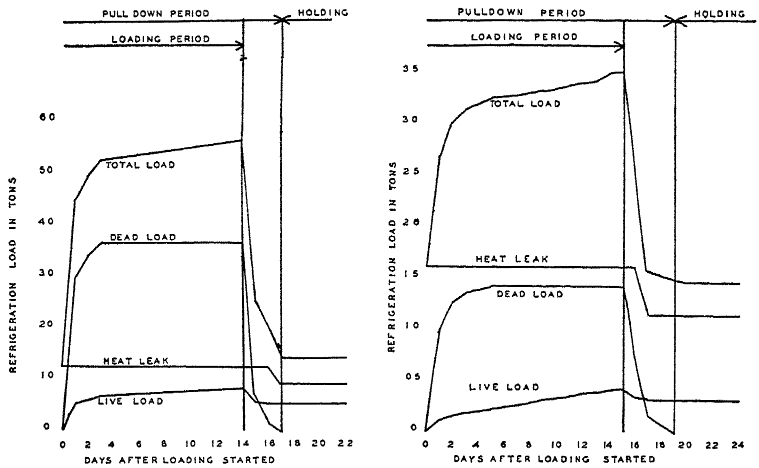

Fig. 12.—Graphic representation of the load on a small (left); and a large (right) apple storage refrigeration system

In table 14 are given the tonnages of refrigeration calculated to cool various-sized storages at various loading rates. It would appear most economical to design any apple storage refrigeration system on the basis of the absolute minimum loading rate per day consistent with the contemplated rate of harvest of various varieties. In addition, it is frequently possible to withhold some of any day's harvested fruit from storage, either by leaving it in the orchard overnight or by temporary storage in the shade or in the packing shed. However, since harvested apples should be reduced in temperature as rapidly

TABLE 14.—Refrigeration requirements for several storages and loading rates*

Storage capacity	Loading rate of 5 per cent	Refrigeration required	Loading rate of 7 per cent	Refrigeration required	Loading rate of 10 per cent	Refrigeration required
Bushels	*Bushels*	*Tons*	*Bushels*	*Tons*	*Bushels*	*Tons*
2,500	125	1.6	175	1.9	250	2.4
5,000	250	3.1	350	3.8	500	4.8
7,500	375	4.7	525	5.7	750	7.1
10,000	500	6.3	700	7.6	1,000	9.5
12,500	625	7.8	875	9.4	1,250	11.9
15,000	750	9.4	1,050	11.3	1,500	14.3
20,000	1,000	12.5	1,400	15.1	2,000	19.0
25,000	1,250	15.6	1,750	19.3	2,500	23.8
40,000	2,000	25.0	2,800	30.2	4,000	38.0
50,000	2,500	31.3	3,500	37.8	5,000	47.5

*Data based on approximate heat leakage load of 0 3 ton per 666 square feet of total superficial area of storage per 1,000-bushel capacity, and respiration and sensible heat load in cooling apples from initial temperature of 80° F. to 35° F. in 1 day.

as possible for their good keeping, it is wise to place all fruit in storage, even at the risk of raising the temperature of fruit already in storage. Loading must be controlled in such a way that the daily heat of respiration plus the daily dead load does not increase beyond the capacity of the refrigeration system to absorb. The extent of such increase in heat load is discussed in the section on live heat load.

Leaving fruit in crates in the orchard overnight is better practice than stacking the fruit in a shed or other shaded place where less free circulation of air will occur. However, gravity air cooling of fruit is very slow, and it is doubtful whether fruit left in the orchard will cool more than 3° F. under the best conditions encountered at harvest time. The cost of rehandling the fruit precludes the practice of leaving the fruit in the orchard unless it is normal to pick and stack the fruit under the trees while waiting for the truck to remove it to the packing shed. A preferable method of handling fruit prior to placing in cold storage would be to place it in temporary storage where forced-air circulation could be applied and more rapid extraction of heat from the fruit during the cool night and morning hours would occur. The main objective in any case should be to keep the fruit shaded at all times, for the direct rays of the sun will increase the temperature of the fruit very rapidly. For this reason, some cover should be provided for all loads of fruit moving some distance from the orchard to the storage in the sunshine.

There is a considerable increase in capacity of any refrigeration system when the temperature of the air passing over the cooling coils increases because of increased fruit load or other cause, and if the refrigeration condenser unit has sufficient capacity to maintain the original suction pressure in the system, a considerably increased cooling effect is obtained. In table 15, this relation between refrigerant temperature (suction pressure), room temperature, and capacity of the refrigeration unit is illustrated.

TABLE 15.—Increase in capacity of a refrigeration unit with rise in room temperature

Refrigerant temperature, degrees F.	Room temperature, degrees F.	Relative capacity
25.4	35	1.00
25.4	40	1.78
25.4	45	2.56
25.4	50	3.34

For all calculations on "dead load" for any storage, it is necessary to assume that each day's loading will be reduced from field to storage design temperature in 24 hours, even though under some conditions the fruit will not reach room temperature for 48 or 72 hours. The rate of pulldown does not affect the dead load calculations; there is a definite quantity of heat to remove for every degree of lowering of fruit temperature. After the first few days of loading, that is, after the first day's loading has been reduced to room temperature, there is an accumulative daily dead load equivalent to removing the sensible heat from each day's loading in 24 hours. This lasts until the last day of loading. If less cooling capacity is supplied, the system would lose ground instead of keeping up with the refrigeration load. The refrigeration equipment must have sufficient capacity to lower the temperature of the daily fruit load to design room temperature so that its full capacity may be available to repeat the cooling process for the second day's loading, and so on, to full loading. Additional capacity is then provided to handle the "live", or respiration, heat load. The higher the design pulldown room temperature, the greater will be the "live" load each day, and for this reason, an attempt is made to set the design room temperature sufficiently low that the daily "live" load is kept at a safe level. Once the fruit temperature is reduced, the dead load disappears and does not appear in the holding period load calculations.

LIVE HEAT LOAD CALCULATIONS

It is well known that apples are a source of heat, especially when held at relatively high temperatures by the chemical processes constantly taking place within the living fruit. This "heat of respiration", as it is commonly called, increases with the temperature. There is a double advantage in maintaining fruit at a relatively low temperature, since the slower respiration consumes less of the apple, and it remains firm and of good eating quality longer, and at the same time, slower respiration requires less refrigeration, for there is less respiration heat to absorb at the lower temperatures.

The heat of respiration, or live load, of apples is extremely variable and depends upon the variety, growing conditions, and the like. In table 16 are given the approximate rates of evolution of heat by apples stored at various temperatures. The data in table 16 were compiled and estimated from original data given by Rose et al. (21). Since the live load may vary between wide limits, it is convenient to select estimated values for storage temperatures of 30, 32, 35, 40, and 80° F., of 500, 700, 1,000, 1,500, and 10,000 B. T. U. per ton per 24 hours, respectively.

TABLE 16.—Approximate rate of respiration and evolution of heat by apples at various temperatures

Temperature, degrees F.	Milligrams of carbon dioxide per kilogram-hour	B.T.U. per ton per 24 hours	Tons of refrigeration per 1,000 bushels
30–32	1– 3	220– 660	0.017–0.052
38–40	4– 7	880– 1,540	.069– .120
45–47	8–13	1,760– 2,860	.138– .223
50–52	12–21	2,640– 4,620	.206– .361
55–57	16–26	3,520– 5,720	.275– .447
60–62	20–33	4,400– 7,260	.344– .567
65–67	22–41	4,840– 9,020	.378– .705
70–72	24–49	5,280–10,780	.413– .842
75–77	26–57	5,700–12,540	.445– .980
80–82	28–65	6,160–14,300	.481–1.117
85–87	30–73	6,600–16,060	.516–1.255

There are two methods of calculating the live load during the pulldown period. Which one to use depends upon whether the incoming fruit is reduced to design temperature in 24 hours, or whether a longer period of 48 to 72 hours elapses before each day's loading reaches design temperature. Systems employing gravity circulation probably never reduce the daily fruit load to design temperature in less than 4 or 5 days. If the fruit temperature is to be lowered rapidly, adequate refrigeration capacity must be provided, and it must be of the forced-air type; the fruit must be stored in crates; and plenty of room must be left around the load and between tiers of crates so that air can move freely over each apple. Storage operators seldom realize that if heat is to be transferred rapidly from the fruit to the refrigerating system, circulating air must pass over each individual apple, and that if this is not possible, adequate system capacity is of no avail. The tendency is to crowd in as much fruit as possible in a given space, and records show that when this is done, the flesh temperature of the apple may remain as high as 10° F. above the air temperature for long periods. Large aisleways are of no advantage, since they act as free passageways for air to return quickly to the cooling unit without picking up much heat. Fruit should be so distributed in a storage that the minimum of large passageways exists, and it is much better to arrange the crates to occupy as much of the room space as possible when the storage is not completely filled. It has been shown (17) that floor racks cause two and one-half times more air movement through the loads than takes place where no floor racks are used. The use of floor racks, dunnage strips, and open containers, and the proper discharge and return of air to the cooling units are essential to 24-hour pulldown of fruit in any storage.

Since it is tedious and rather unnecessary to calculate the daily live load for each day as the storage is loaded, and since fruit does not always enter storage at a uniform rate, it is sufficient to calculate the peak live load during pulldown by using the average between the mean respiration rate per ton per day at entering fruit temperature, and the mean rate at storage temperature. For example, if the fruit entered at 80° F. and the storage temperature was 35° F., the mean respiration per ton per day would be $\frac{10{,}000 + 1{,}000}{2}$, or 5,500 B. T. U. Assuming that the 10,000 bushels are stored at the rate of 500 bushels per day, the live load due to 1 day's loading would be:

$$\frac{500 \text{ bushels} \times 50 \text{ pounds per bushel} \times \frac{10{,}000 + 1{,}000}{2}}{2{,}000 \text{ pounds per ton}} = 41{,}250 \text{ B. T. U.}$$

The live load due to fruit already at storage temperature would be calculated in the same manner. At the last day of loading, there would be 10,000—500, or 9,500 bushels, of fruit in storage at 35° F. with a mean respiration rate per ton per day of 1,000 B. T. U. Therefore,

$$\frac{9{,}500 \text{ bushels} \times 50 \text{ pounds per bushel} \times 1{,}000}{2{,}000 \text{ pounds per ton}} =$$

237,500 B. T. U. per day, live load due to fruit already at storage temperature. If each day's loading was cooled to room temperature in 24 hours, the total peak live load would be the sum of the live load for the 500 bushels loaded per day (41,250 B. T. U.) plus the live load from the fruit already in storage (237,500), or 278,750 B. T. U. per day.

When conditions are not favorable for reducing the temperature of each day's loading to design temperature in 24 hours, the increased respiration rate due to the delayed pulldown must be taken into consideration. This condition requires that the average of the mean live load as calculated be determined for the fruit entering storage for the first 24-hour period and the second and the third days, after which each day's loading will probably have reached design room temperature. It is hardly necessary to calculate the peak load with such detail as determining each day's live load separately and accumulating them until peak load is reached. A simpler method is to employ the mean respiration per ton per day at storage temperature and a respiration factor which takes into consideration the increased respiration rate due to the delayed pulldown. When a respiration factor is applied to the example given, the equation is as follows:

$$\frac{10{,}000 \text{ bushels} \times 50 \text{ pounds} \times 1{,}000 \times 1.5, \text{ respiration factor}}{2{,}000 \text{ pounds}} = 375{,}000 \text{ B. T. U.}$$

per ton per day, peak live load during pulldown period.

When the peak live load calculated on the basis of a 24-hour pulldown (278,750 B. T. U.) is compared with the peak live load calculated on the basis of a delayed pulldown (375,000 B. T. U.), the difference (96,250 B. T. U. per day) represents the additional daily peak load on the refrigeration system which must be absorbed in order to prevent the room temperature from rising, and eventually to lower the total fruit load to design room temperature. This difference shows why it is economical to size the equipment properly and to make all effort to lower each day's loading to room temperature in 24 hours.

The total cooling load per thousand bushels of fruit can readily be obtained from table 17. The respiration load from fruit already in storage is not included and must be calculated separately.

TABLE 17.—Approximate refrigeration load in tons per 1,000 bushels to cool apples from several temperatures to 35° F. in 1 to 10 days*

Initial temperature, degrees F.	Respiration Days in cooling										Sensible
	1	2	3	4	5	6	7	8	9	10	
85	0.21	0.43	0.64	0.86	1.07	1.29	1.50	1.72	1.93	2.15	7.03
80	.21	.42	.63	.84	1.05	1.27	1.48	1.69	1.90	2.11	6.33
75	.20	.41	.61	.81	1.02	1.22	1.42	1.62	1.83	2.03	5.63
70	.19	.38	.57	.77	.96	1.15	1.34	1.53	1.72	1.91	4.92
65	.18	.35	.83	.70	.88	1.05	1.23	1.41	1.58	1.76	4.22
60	.16	.31	.47	.63	.78	.94	1.09	1.25	1.41	1.56	3.52
55	.13	.27	.40	.53	.66	.80	.93	1.06	1.20	1.33	2.81
50	.11	.21	.32	.42	.53	.63	.74	.84	.95	1.05	2.11
45	.07	.15	.22	.30	.37	.45	.52	.59	.67	.74	1.41
40	.04	.08	.12	.16	.20	.23	.27	.31	.35	.39	.70

*Compiled from data of Rose et al. (21).

In table 17 are given the approximate refrigeration loads in tons per 1,000 bushels to cool apples from any one of several initial temperatures to 35° F. in 1 to 10 days. The "sensible" heat load, or the heat extracted in lowering the temperature of fruit, varies only with the cooling range, and not with the time required to cool that fruit, and is given in the last column of table 17. The "live", or respiration cooling, load varies with time, and for all practical purposes can be considered cumulative and to be in proportion to the time required for cooling the fruit to the desired storage room temperature. The sum of the "respiration" heat and "sensible" heat loads is the total cooling load.

HOLDING HEAT LEAK AND LIVE LOAD CALCULATIONS

The total cooling load on the refrigeration system during the holding period, after all the fruit has been reduced in temperature to the design room temperature, is made up of the heat leak and the live loads. It is obvious that it will be much less than the total pulldown load and, therefore, a refrigeration system which can handle the pulldown load will have more than enough capacity to handle the load during the majority of the storage season. In order to keep the investment in refrigeration equipment as low as possible, particularly since most of the equipment capacity is used only a very few days out of the year, it is wise practice to load the storage as slowly as is consistent with good keeping of the fruit and to favor the equipment by loading at a uniform rate. In this way, the grower can avoid an increased investment in refrigeration to handle a very temporary situation, that is, lowering the temperature of incoming fruit during a period as short as 15 days.

There is one good reason for calculating the holding load. If he knows it, the storage owner can select compressor equipment in two units properly sized so that one machine can be shut down completely and the remaining smaller machine allowed to run 18 to 24 hours to carry the lighter holding load. In this way, some economy can be secured by lowering operating and maintenance costs through the operation of one small machine continuously rather than one large machine less frequently.

The holding heat leak load can be determined from the pulldown heat leak load by determining the ratio of the holding heat leak differential to the pulldown heat leak differential and applying this ratio to the pulldown heat load. In the sample storage described, the holding heat leak differential is the difference between the expected outdoor temperature during the holding period and the design storage temperature, which in this example is 60° minus 35° F., or 25° F. The differential for the pulldown period is 80° minus 35° F., or 45° F. The ratio 25/45 multiplied by the pulldown heat leak gives the holding heat leak load. The holding live load is determined, as already explained, by using the mean respiration rate at holding temperature.

The total pulldown and holding loads can be determined by adding together the heat leak, dead load, and live load. From these totals, the number of condensing units and the capacities necessary can easily be determined. From such calculations, the refrigeration company salesman determines which units of his line will best handle the varying cooling loads for any particular storage. He must decide whether the condensers will handle the load operating 18 hours out of 24, or more. The decision in any case depends upon the design standards being followed. For example, for automatic defrosting, which can be obtained on a design room temperature of 35° or above, 18-hour running time is preferable for gravity air circulation equipment. Twenty-four-hour operation will be suitable for determining equipment size for all forced-air units, as they will automatically defrost on 35° F. room air during the holding period and can be defrosted manually when necessary during the pulldown period. Seldom will the equipment operate 24 hours without stopping, since outdoor temperatures frequently are below the peak temperature used in calculations made to select the proper size of equipment. The longer the running time, the better will be the control over room temperature, humidity, and air movement.

It has been pointed out that it is very important to balance the condensing unit and evaporator capacities. When this balance is accomplished, based on a narrow temperature differential across the cooling coils or evaporator, major emphasis is being placed on "holding-period" operation. With the same equipment, it is permissible that the temperature differential become greater during the pulldown period, for moisture control is not so important during the time of loading as it is later. The warm apples entering storage supply the necessary moisture to maintain a relatively high humidity, and very little of this moisture comes from the fruit already cooled to room temperature. In addition, the wide temperature differential existing during loading increases the capacity of the refrigeration system when it is greatly needed. It is with this method of operation, in which long running time for the equipment is most desirable, that two or more compressors meet the requirements best. Two-speed motors, various-sized pulleys, or compressor capacity reducers also provide a flexibility in refrigeration capacity which is an advantage in maintaining ideal conditions in the storage.

RESULTS OF TESTS ON STORING APPLES

STORAGE TESTS BEFORE 1936

The Ohio Agricultural Experiment Station has been storing apples and examining them for storage disorders for more than 10 years. Both common and refrigerated storage rooms have been used in the tests. One of the objects of securing such data, to determine the effect of orchard cultural practice, including the application of fertilizers, on the keeping of the fruit, has been reported by Gourley and Hopkins (12). In these storage tests, various methods of packing were employed. Fruit of varying size was harvested at successive dates in the fall. From the data thus secured is obtained some knowledge of how fruit should be harvested and packed for long storage life.

In tables 18, 19, and 20 are given portions of the data secured to show the distribution of the various storage disorders within seasons and varieties. Three-bushel samples of fruit were taken from several trees in each plot, in order to represent the total quantity of fruit harvested as closely as possible, and the samples, with the exception of those specifically mentioned, were packed in shredded oiled paper in bushel baskets with caps and lids. Each month the fruit was removed and examined for disorders. These examinations were carried on until April, and the long test period accounts for the high percentage of some disorders certain seasons. This work brought out the strong effect of type of growing season on the appearance of such disorders, and the wide variation due to season is shown in table 18, which shows that Jonathan "spot" and Baldwin "spot" were nonexistent during 1928 and very slight during 1935 and 1937.

TABLE 18.—Per cent of fruits in storage affected with Jonathan spot and Baldwin spot from 1928 to 1937 inclusive

Year	Jonathan	Baldwin
1928	0	0
1929	88	35
1930	72	28
1931	58	15
1932	33	21
1933	44	24
1934	23	33
1935	0	3
1936	30	55
1937	0	7

TABLE 19.—**Internal breakdown on Jonathan, Stayman Winesap, Grimes Golden, and Baldwin**

Per cent of fruits in storage affected

Year	Jonathan	Stayman Winesap	Grimes Golden	Baldwin
1929	3	0	5	0
1930	7	10	0	1
1931	1	5	3	10
1932	20	5	0	6
1933	0	2	0	3

TABLE 20.—**Scald on Grimes Golden, Stayman Winesap, Baldwin, and Jonathan**

Per cent of fruits in storage affected

Year	Grimes Golden	Stayman Winesap	Baldwin	Jonathan
1928	23	14	0	7
1929	61	7	3	0
1930	69	0	0	0
1931	42	12	2	0
1932	68	8	6	3
1933	80	30	5	0
1934	42	0	....	....

The effect of date of picking and storing and the method of wrapping on the occurrence of shriveling and scald in 1929 is shown in table 21. In almost all the samples of Golden Delicious, the fruit harvested and stored on October 13 showed less shrivel in both basket and crate than samples stored on either October 5 or 18. There was only a slight difference in shriveling between plain and oiled wrappers when the fruit was harvested either early (October 7) or late (October 18), but that packed in shredded oiled paper shriveled considerably less when picked late, regardless of type of storage. There was distinctly less shriveling when the fruit was wrapped in oiled paper or packed in shredded oiled paper than when plain paper was used, or when the fruit was not wrapped (table 21). The type of container had a distinct effect on the appearance of shriveling; the least shriveling occurred on the fruit held in baskets. There was slightly, but consistently, less shrivel in the refrigerated room than in the cellar room during this single season.

The data in table 22 are of interest primarily because they indicate that Baldwin spot is much more prevalent in refrigerated, than in common, storage. Although the occurrence of the other storage disorders was not of great magnitude, the data show their occurrence, with the exception of shriveling and Baldwin spot, to be more frequent in the common storage.

STORAGE TESTS SINCE 1936

Since 1936, the Ohio Station has carried on more extensive studies on the keeping of apples in storage. In 1936, a modern refrigeration plant was installed in a common storage. The equipment included a 3-horsepower Freon compressor connected to a fin-type forced-air cooling unit. Spray-type water nozzles (with water pressure only) were installed to increase the relative

TABLE 21.—Effect of method of packing Golden Delicious apples on occurrence of shriveling and scald, 1929

Stored October—	Refrigerated storage 5		13		18		7		18	
	Basket	Crate	Basket	Crate	Basket	Crate	Refrigerated Basket	Cellar Basket	Refrigerated Basket	Cellar Basket
	Per cent of shriveling									
No wrapping	80	98	61	89	70	98	....	....	....	....
Plain paper wrapping	86	97	70	94	77	91	67	73	75	79
Oil paper wrapping	62	89	41	76	56	95	48	56	57	49
Shredded oil paper	64	38	36	47	42	31	43	52	16	29
	Per cent of scald									
No wrapping	12	0	35	11	19	0	No counts made on scalded fruit			
Plain paper wrapping	9	1	10	5	14	7				
Oil paper wrapping	6	0	1	0	0	0				
Shredded oil paper	9	2	1	2	2	3				

TABLE 22.—Effect of size, color, and season on storage disorders with Baldwin

Per cent of fruits affected, average of 10 one-tree plots

Stored—	Large 3 inches and above		Medium, below 3 inches		Highly colored medium to large	Widest range within 10 trees	
Type of storage—	1934 Refrigerated	1935 Common	1934 Refrigerated	1935 Common	1935* Common	1934 Refrigerated	1935 Common
Decay	0	5	0	3	3	0-2	0-10
Breakdown	1	9	0	4	7	0-3	2-20
Shrivel	0	1	2	1	0	0-4	0-3
Scald	0	1	0	0	0	0	0-4
Baldwin spot	51	6	21	2	1	31-75	0-10

*Data for refrigerated storage were not obtained in 1934.

humidity in the storage room. The temperatures and relative humidities maintained in these storages during two seasons are shown in figures 13, 14, 15, and 16. The underground storage has been described in detail elsewhere (7). The aboveground air-cooled room was insulated with reflective-type insulation and equipped with power fans to control temperatures (8). The direct-expansion ammonia-cooled room was used for many years and represents the older type of cold storage installations in which the air circulates slowly by gravity from coils at the ceiling to the floor and back to the coils.

It was the aim of these tests to determine how apples packed in various types of containers keep under the various types of storage conditions. The graphs in figure 13 show that during 1937-1938, the air temperature was very constant near 35° F., the thermostat setting in the blower-equipped storage. A short period during the time the storage was being loaded, the temperature fluctuated somewhat above the thermostat setting. By means of high back pressure at the compressor, the refrigerant temperature was maintained relatively high, and the storage air circulated over the coils was cooled only slightly. This operation maintained a higher relative humidity of the delivered

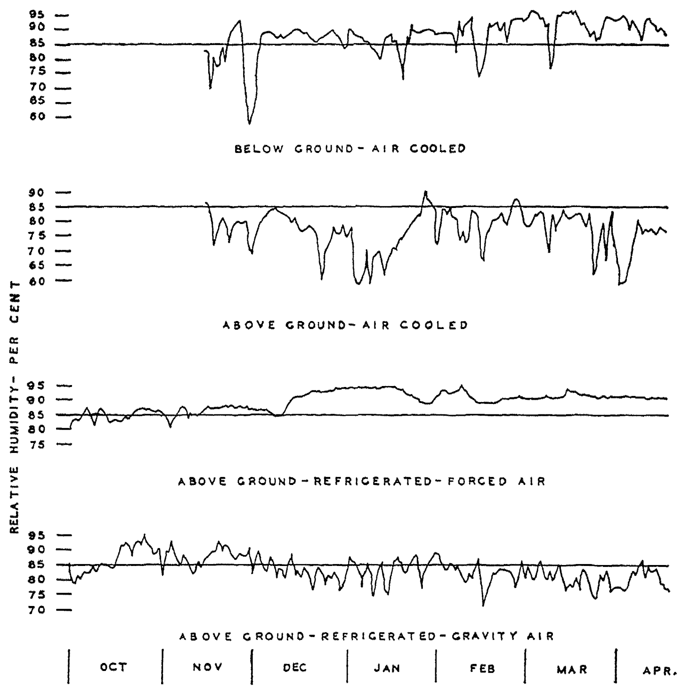

Fig. 13.—Daily mean air temperatures outside and in four types of storages, 1937-1938

air, as has been explained elsewhere. Figure 14 shows that this humidity averaged 85 per cent or better until the middle of December, when the water spray nozzles were turned on, and the humidity was boosted to near 90 per cent relative for the remainder of the storage period.

In the other refrigerated storage, with the gravity air movement, for the same period (fig. 13 and 14) the air temperature fluctuated considerably from day to day because of poor operation and control of the refrigeration equipment. However, the mean temperature maintained is of first importance, and this was close to, or slightly lower than, that maintained in the blower-equipped room. The humidity was also much more variable and averaged nearer 80 per cent relative during most of the season. This condition was unavoidable and is characteristic of the bare-pipe coil using ammonia at temperatures of 5 to 15° F., for at these low temperatures, it has a great tendency to condense more moisture from the air and lower the humidity, as has been explained.

The graphs for the storage seasons of 1938-1939, presented in figures 15 and 16, show that the room temperature in the blower storage was more variable than for the previous year, and at a somewhat lower average. During the season, provision was made to circulate outside air through the storage by means of the blower fans and thus reduce the total power load. This practice allowed a greater variation in room temperature, which was compensated for

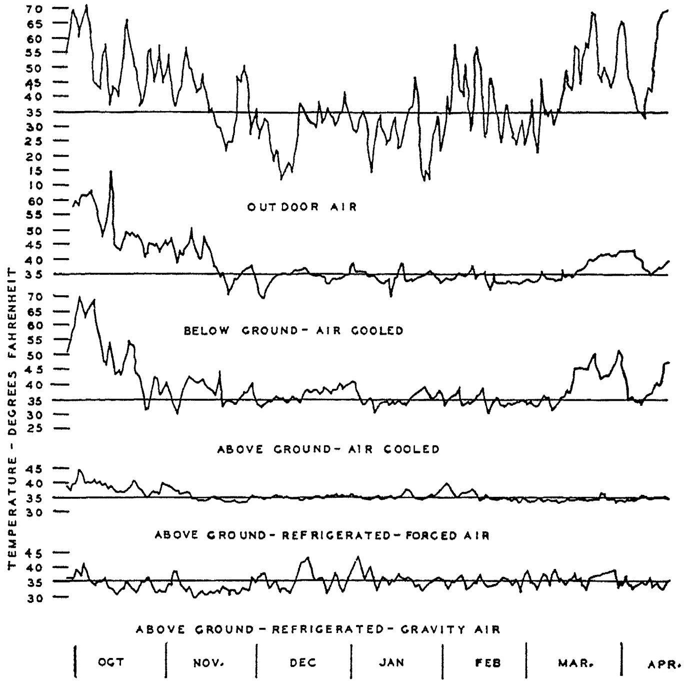

Fig. 14.—Daily mean relative humidity in four types of storages, 1937-1938

by a lower mean temperature. The temperature in the room cooled with the bare-pipe coil was somewhat more constant and slightly lower than during the previous season.

The relative humidity averaged lower in both storages during the second season and fluctuated to as low as 70 per cent in the room cooled with forced air. Although the water spray nozzles were operated continuously when the storage was cooled with outside air, it was impossible with the equipment available to maintain a high relative humidity all the time, and only when the room was closed, was the humidity satisfactory. This phase of the problem of forced-air cooling has already been discussed.

In order to answer the question as to whether apples will keep as well under forced-air cooling as under the older, bare-pipe coil type of refrigeration, and to determine the factors responsible for any differences in keeping of fruit stored under widely differing conditions, several varieties of apples were stored in baskets, boxes, and crates in several positions in four types of storages. The storages employed in these studies have already been described. Care was taken to choose representative fruits of each variety, and several bushels of each were moved into storage the day they were harvested. Smaller samples were selected for special studies, such as weight loss, total soluble solids, titratable acids, moisture percentages, and firmness of flesh.

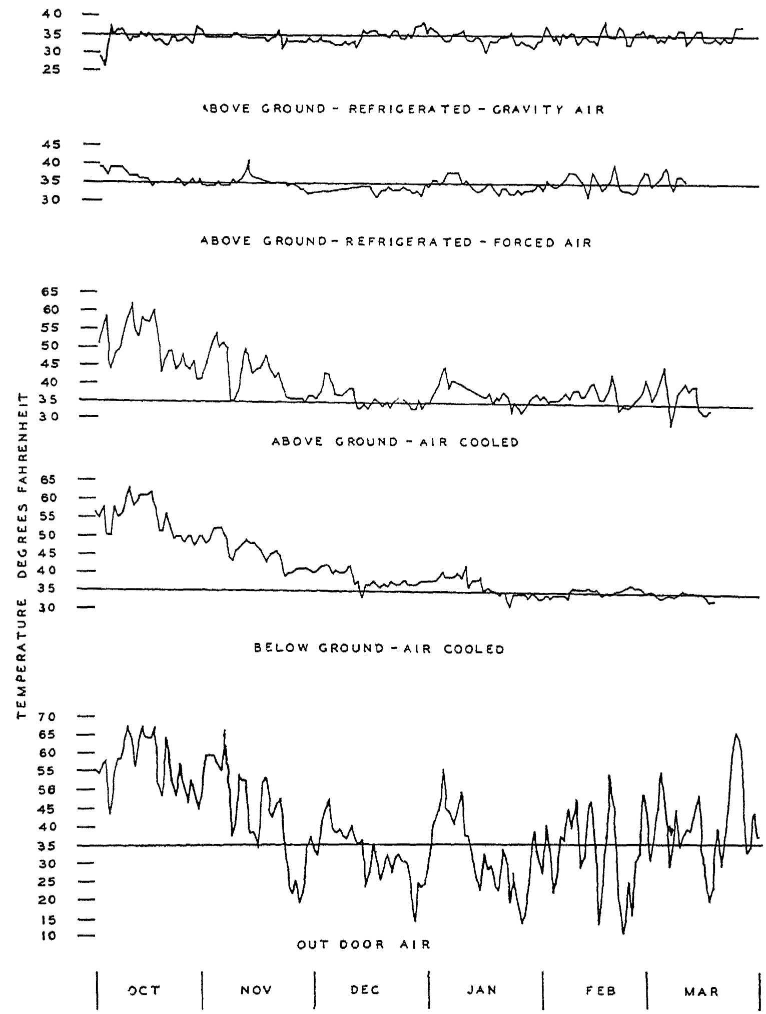

Fig. 15.—Daily mean air temperatures outside and in four types of storages, 1938-1939

Only those data of value in showing differences in the actual condition of the fruit at the completion of the storage period are presented. The percentage of titratable acid determined each month on Baldwin fruits from each storage varied between 1.1 and 2.4, with no significant differences observable between any of the storages. Consistent differences did occur from month to month. Likewise, the total soluble solids as determined by means of an Abbey refractometer ranged from 10 to 14.9 per cent, depending on the time the sample was taken, but no consistent differences were noted between treatments.

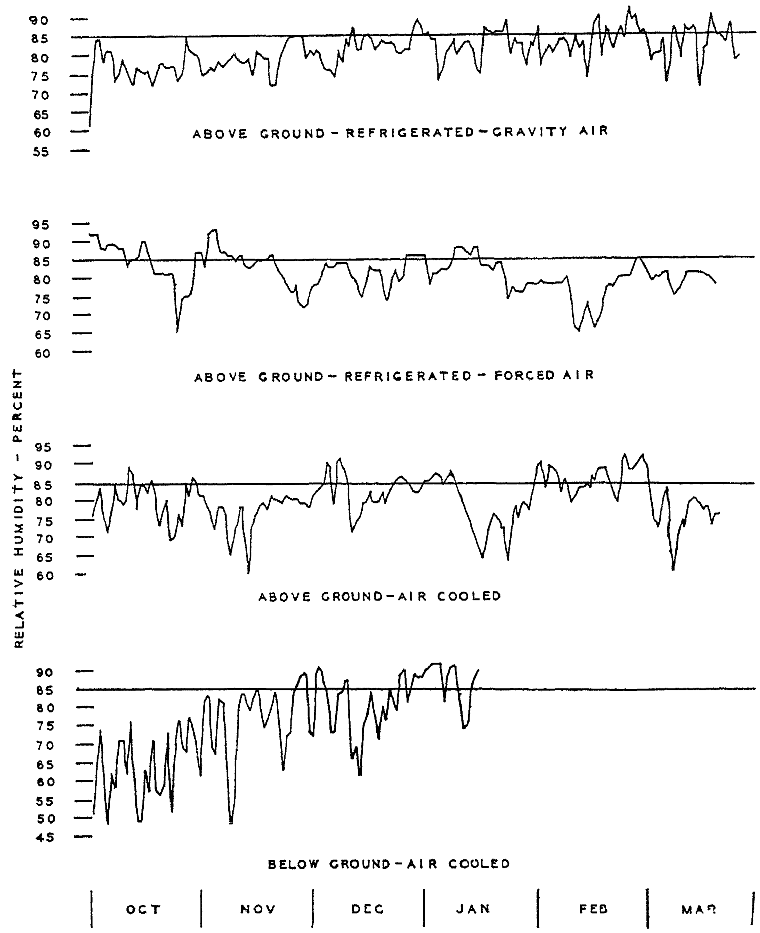

Fig. 16.—Daily mean relative humidity in four types of storages, 1938-1939

Although the apples lost considerable total moisture (loss in weight) during the storage period, the percentage of water in the fruits remained constant. This result indicates that the sugars and other carbohydrates in the fruits were lost in respiration at somewhat the same rate as the moisture. For example, the water in fruits ranged from 71 to 72 per cent of fresh weight after 1 month in storage and 67 to 70 per cent at the end of the sixth month.

The daily mean relative humidity occurring in four different types of storages during two seasons has been given in figures 14 and 16. A second method of determining the "pulling" power of the air for moisture is by means of porous clay bulbs commonly known as atmometers (9). These instruments, through suitable connection to measurable volumes of water, indicate the evaporation loss from the porous clay surface and thus show the evaporative power of the air. Air movement is known to increase evaporation under most conditions, and the atmometer affords a measure of the "pulling" power of the air for moisture which is not indicated in relative humidity measurements.

In table 23 is given the evaporation in milliliters per square inch of clay surface per week in four storages for two seasons. The evaporation is considerably higher during the loading period in the two air-cooled rooms, probably because of the higher temperatures existing in these two storages, as well as greater movement of dry outside air through the rooms. The greatest fluctuation in evaporation occurred in the underground storage, as would be expected, since the relative humidity rises, and air movement practically ceases, when an underground room is closed. The temperature of the air also drops in the early fall, when the ventilators are closed, and this drop reduces evaporation still further. During most of the storage period for both seasons, the refrigerated room with air circulated by gravity exhibited the highest evaporation of any of the rooms. The relative humidity followed the same course, and thus all data indicate that such a system of refrigeration causes more water loss from fruit, particularly in the later period of storage, than do other types.

TABLE 23.—Water evaporated from porous clay surface in four storages*

Evaporation in milliliters per square inch per week

Date week ended	Storages							
	Refrigerated forced-air		Refrigerated gravity air		Air-cooled aboveground		Air-cooled belowground	
	1937–1938	1938–1939	1937–1938	1938–1939	1937–1938	1938–1939	1937–1938	1938–1939
November 1	89	126	65	133	156	272	138	170
8	95	133	65	159	211	232	158	119
15	82	146	117	133	187	239	168	125
22	70	139	71	146	100	232	142	218
29	51	113	71	146	69	159	135	73
December 6	76	115	104	126	69	146	100	60
13	70	113	97	139	31	173	98	53
20	63	93	136	113	31	154	22	113
27	57	100	100	203	38	133	14	60
January 3	44	79	123	139	25	86	15	20
10	57	59	87	139	69	113	28	20
17	51	93	97	146	69	133	30	40
24	63	100	102	146	63	118	29	27
31	38	100	112	154	44	66	25	13
February 7	70	86	136	154	56	64	27	15
14	57	113	220	154	13	66	22	12
21	51	106	112	133	56	67	79	10
28	38	93	117	133	19	72	22	12
March 7	44	79	95	167	31	78	22	18
14	63	118	103	139	50	65	14	27
21	63	93	123	154	38	79	14	26
28	63	94	117	148	88	68	14	28

*Atmometer bulbs used as evaporating surfaces.

The forced-air refrigerated room held as high a relative humidity and as low an evaporation throughout the two seasons as any of the storages, with the exception of the belowground type, which after the first of December, when it was seldom opened, maintained a very high relative humidity and low evaporation rate. Moisture was added to the air of the forced-air refrigerated room by means of nozzles from time to time, and during 1938-1939, outside air was used for cooling when conditions were favorable. This result is of considerable interest, since many growers are of the opinion that forced-air equipment will cause low relative humidity and high evaporation in storages. Atmometers placed in various locations throughout the storage rooms did show considerable variation in evaporation rates, primarily correlated with air movement. A comparison of the evaporation index in the storages by weeks is of more value than any mean value for a longer period.

The apples were examined to determine the importance of humidity and evaporation. Apples of several varieties were stored in crates, boxes, and baskets, and weighings were made each month. The loss in weight was considered to represent the moisture loss from the fruit, although some small portion of the weight loss was due to respiration. The test was made in quadruplicate, and weighings were continued to well beyond the normal sale time for the apple during 1938-1939 to accentuate the effect of the storage atmosphere on moisture loss. Earlier tests with small numbers of fruits had shown that moisture loss was slow at first and that the slight differences between varieties, containers, and position in storage were not significant when a small numbers of samples was used. Therefore, only the accumulative weight losses are given in tables 24 and 25. The data for 1937-1938, secured over a period of only 5 months, show smaller percentage losses, and they are more uniform than those taken during 1938-1939. Table 24 shows slight differences in moisture loss from fruit in the four types of storages and that the least loss was in the blower room and the greatest in the coil room. The difference between storages and between containers was not significant during the relatively short period in which moisture losses were measured. Table 25 shows data secured over a 7-month period and greater differences. Again the room cooled with a bare-pipe coil without a blower showed a slightly greater loss of moisture than the blower-equipped room. The loss in weight of the three varieties in percentage of original weight was greatest in the crate, followed, in order, by the box and basket. The position of the fruit in storage seemed to have considerable effect on moisture loss. As long as the relative humidity was maintained at 85 per cent or above, the weight loss of the apples was slow and not a factor of importance up to the time of sale of most varieties.

TABLE 24.—Loss in weight (in per cent) of apples stored in different storages

Average of seven varieties and four replicates, October 1937 to February 1938

Storage container	Type of storage			
	Aboveground equipped with—		Underground	Aboveground
	Blower and coil	Coil only		
Crate	2.5	3 1	2 5	2.9
Box	2.5	3.1	3 2	2.7
Basket	2.5	3.0	2.8	2.5
Average of three containers	2.5	3.1	2.9	2.7

Apples are seldom sold by weight, and thus losses in weight up to 5 per cent, where visible shriveling commences, can be ignored. However, their appearance does affect sales, and, therefore, weight losses must be kept below the point at which visible shriveling occurs. Moisture loss control cannot be ignored, especially when the apples are held for more than 5 months. The blower type of equipment with proper coil temperature control can be relied upon to provide the minimum requirements for humidity control.

TABLE 25.—Loss in weight (in per cent) of apples during storage period, October 1938 to April 1939

Average of four replicates

Storage container	Type of storage			
	Aboveground equipped with—		Underground	Aboveground
	Blower and coil	Coil only		
	Delicious			
Crate	6.0	8.5	9.0	3.5
Box	5.5	6.0	4.5	3.0
Basket	4.0	4.0	2.5	3.0
Mean	5.2	6.2	5.3	3.2
	Baldwin			
Crate	8.0	9.0	5.5	5.5
Box	7.5	7.5	5.5	5.5
Basket	6.5	4.0	5.5	4.5
Mean	7.3	6.8	5.5	5.2
	Stayman Winesap			
Crate	5.0	8.0	5.0	6.0
Box	4.5	4.5	3.0	3.5
Basket	4.5	4.5	5.0	3.5
Mean	4.7	5.0	4.3	4.3
Mean of three varieties	5.7	6.0	5.0	4.2

Different lots of fruit were examined monthly for shriveling, and through recording the percentage of apples so affected, it was possible to determine at what period in their storage life weight losses became serious enough to affect the sale of the fruit. In table 26 is given a summary of the disorders found on inspecting the test lots each month while in storage. The forced-air refrigerated storage held fruit as well as, or better than, the gravity air refrigerated room, especially when scald and shriveling alone are considered. Shriveling was least in baskets, but scald was greatest in these containers, and for this reason, the open container, such as the crate, is recommended where a relatively high humidity can be maintained.

The data in table 26 indicate that shriveling is the most important storage disorder with which the grower must contend, and that scald is next. Other disorders, such as decay, Jonathan and Baldwin spot, and breakdown are not of much importance except during certain seasons.

Shrivel and scald increase with storage time but not at a uniform rate. After the fruit has been in storage 100 days, these disorders may increase more slowly.

TABLE 26.—The effect of several factors on the keeping of apples in storage

Type of storage	Fruits examined		Per cent of fruit affected*							
			Shrivel		Scald		Decay		Spot†	
	1937	1938	1937	1938	1937	1938	1937	1938	1937	1938
Effect of storage										
Forced air, refrigerated	11,900	4,814	18	25	17	.....	3	.....	2	8
Gravity air, refrigerated	8,134	4,997	34	33	18	.....	2	1	3	8
Common, aboveground	12,739	4,927	32	28	18	.....	.....	4	2	15
Common, belowground	12,212	5,126	30	10	12	2	1	6	1	8
Effect of container										
Type of container:										
Crates	15,493	6,535	42	30	9	1	2	4	2	11
Boxes	15,048	6,819	29	25	16	.....	1	2	2	11
Baskets	14,444	6,508	15	18	24	1	2	3	3	7
Effect of time in storage										
Days in storage, approximate:										
75	44,985	19,862	7	8	.....	.....	.....	1	.....	7
100	.....	.....	14	13	4	.....	.....	1	1	8
150	.....	.....	20	18	7	.....	.....	2	1	8
175	.....	.....	21	24	11	.....	.....	3	1	10
200‡	.....	.....	26	.....	13	.....	1	.....	1	.....
Effect of variety										
Variety stored:§										
Stayman Winesap	9,249	6,789	9	32	14	1	1	2	.....	5
Baldwin	6,868	6,349	18	36	.....	.....	1	4	2	3
Delicious	6,707	6,724	16	6	.....	.....	1	3	.....	.....
Jonathan	9,494	.....	55	.....	.....	.....	2	.....	8	.....
Grimes Golden	9,248	.....	34	.....	60	.....	3	.....	.....	.....
Golden Delicious	1,545	.....	76	.....	1	.....	3	.....	.....	.....
Rome Beauty	1,874	.....	.....	.....	16	.....	1	.....	.....	.....

*Table includes fruits affected to the extent of 1 per cent or more.
†Refers to spot commonly found on Jonathan and Baldwin.
‡Fruits not examined at this time in 1938.
§The first three varieties only were stored in 1938. Golden Delicious and Rome were packed in crates only.

The amount of any disorder appearing in storage, including shrivel, is dependent primarily upon the apple variety. This condition is unfortunate, since all varieties are stored together in most Ohio storages. It is recommended that all apples be stored in crates, or with oiled paper or wraps, when stored in baskets, and that the relative humidity be held as high as possible to prevent shrivel. The higher the storage temperature, the higher the relative humidity required to reduce shriveling. Eighty-eight per cent relative is not too high for storage temperatures of 35° F. or above, and 85 per cent relative should suffice for temperatures below 35° F. Special care in humidity control is necessary when such thin-skinned varieties as Jonathan, Grimes Golden, and Golden Delicious are stored in quantity. The forced-air type of refrigeration equipment can aid in maintaining high relative humidities, in securing good ventilation, and in rapid cooling of the fruit, which are all vital to the keeping of apples in good, salable condition for considerable periods.

The problem of easily and inexpensively maintaining high relative humidities of storage air, especially when cooling is done with low-temperature outside air, is still to be solved. Also, some practical control for scald other than oiled paper is urgently needed.

DISCUSSION AND SUMMARY

Many fruitgrowers, because of necessity and preference, as well as economy, are now installing or contemplating the installation of refrigeration at their farms. The cost of these refrigerated farm storages approximates $1.00 per bushel of capacity. Refrigeration for an existing storage costs from 20 to 50 cents per bushel of capacity to install and close to one-half the total storage cost (17 cents), or 8 cents per bushel per season, to own and operate. This amount is somewhat less than the annual storage cost for apples in commercial storage.

Considerable attention should be given to the location, size, and construction of the building to be refrigerated for apples, since the success of the venture depends largely on these factors. The sales value of a storage-salesroom combination is considerable. The larger storages cost less per bushel of capacity to construct and operate. The loose-fill type of insulation properly installed and combined with masonry supporting walls provides an economical construction for this type of storage. Galvanized metal sheeting for interiors and cement - ground cork or cement-sawdust mixtures for floor insulation aid in lowering costs and lengthening the life of the investment. There are several methods of calculating the proper thickness of insulation and protecting it from moisture in vapor form.

The dry-coil, forced-air, floor-mounted units afford the greatest satisfaction and economy in refrigeration for apples. Both ammonia and Freon serve well as refrigerants, and each has its advantages and disadvantages. Freon is gaining in popularity, especially in the smaller refrigeration systems, those below 10-ton capacity. Ducts for the distribution of the cooled air seem unnecessary and only an added expense. They are used, however, where two or more rooms are cooled by a single unit.

Evaporative or shower-type condensers and cooling towers are proving very popular on farms, since they save up to 95 per cent of the water that otherwise would be needed to condense the refrigerant. It has proved feasible to re-use the water pumped through waste water condensers (shell-and-tube, shell-and-coil, and shell-and-finned tube types) by exhausting it on the storage roof either through ordinary orchard spray nozzles or directly onto the roof, where it is cooled sufficiently for re-use after collection in a large reservoir tank. Small window-mounted evaporative-type condensers used as one or more units are very economical and have the advantage of operating as strictly air-cooled condensers as winter approaches and the air temperature drops sufficiently to allow dispensing with water entirely. By using them, the need for heating the machine room to keep the water pipes from freezing is eliminated.

In selecting cooling units (evaporators) of the finned-tube, forced-air type, it is of the utmost importance to have sufficient total coil area that the maximum daily cooling load can be handled with no more than 8 to 10° F. difference between the refrigerant and air temperatures. A greater differential than 8 or 10° F. will reduce the relative humidity of the delivered air and, at the same time, may cause sufficient frosting of the coils to restrict greatly the movement of storage air across them. Many difficulties in cooling the fruit follow when this mistake in design occurs. The proper design will automatically maintain an 85 per cent relative humidity during the holding period in a reasonably vaportight room.

The most satisfactory method of adding moisture to storage air when it is deemed necessary is through the use of pneumatic water nozzles. By means of a humidistat, small air pump, pneumatic water nozzle, and a water reservoir, the room humidity can be controlled automatically at any desired level with no wetting of the fruit, containers, or surroundings.

The operation of the refrigeration system in farm storages can be made entirely automatic if the system is properly designed, sized, and installed for the cooling load contemplated. Too often, this is not done, to the disappointment of the operator. Some understanding of the capacity and functioning of the unit is necessary so that it can be readjusted from time to time to maintain its most efficient operation.

The holding temperatures for apples in storage are very close to the minimum permissible for automatic defrosting of dry-coil forced-air cooling equipment and for this reason, the frosting and defrosting of the coils are of the greatest importance. Proper sizing of the equipment and the use of properly adjusted controls will eliminate the frosting troubles. Some knowledge of the operation of the controls and cooling equipment is very useful to the operator in maintaining continuous trouble-free refrigeration. The collection of frost on the cooling coils of the direct-expansion bare-pipe coil systems is not nearly so serious as with the blower-equipped units, and for this reason, many growers prefer the former method of cooling their apples, despite the disadvantage of low humidity, slow cooling of fruit, and large amount of space occupied by cooling pipes. By installing wall- or ceiling-mounted fans in conjunction with the latter system, much better performance can be secured.

The accurate calculation of the peak heating load to be handled by a proposed refrigeration system is essential for the proper selection of the equipment. This calculation is most often made by the sales engineer representing the refrigeration machinery company. However, the purchaser of the equipment should understand fully that the peak load is almost entirely dependent upon the rate per day at which the fruit will be unloaded into storage, and that by keeping this figure to a minimum, considerable savings in equipment costs can be made. The rapidity with which the fruit is lowered in temperature is of great importance, since it is imperative that each day's loading of fruit be reduced in temperature to a predetermined room temperature, or else the accumulative load on the cooling equipment will increase beyond the capacity of the system to absorb. The room should be loaded at not more than 10 per cent of its capacity per day, and preferably 5 per cent. Greater daily loadings are permissible when longer intervals between loading are allowed.

In general, the most economical storage is insulated with the equivalent of 4 inches of corkboard on the walls and ceiling and 2 inches in the floor. It is important that the floor insulation should not be ignored, for heat coming in from the floor can increase the yearly power bill for refrigeration out of all proportion to the ratio of the floor to the total superficial area of the storage room.

Any equipment that will handle the peak cooling load at loading time will be more than ample the remainder of the season. For this reason, two compressors are sometimes used so that the lighter cooling load during the holding period can be more economically handled with one small compressor. Cooling with outside air can be resorted to when automatic humidity control with pneumatic water nozzles is supplied, and in this way, still greater economy in operation can be secured during the cold winter months.

The results of storage tests carried on over a period of years with apples emphasized that the seasonal effect was primarily responsible for the appearance of disorders, with the exception of shrivel, which showed no significant difference due to season. The variety was also of great importance in the occurrence of disorders, including shrivel. The effect of storage, container, and time in storage was not so strongly evident. There was more shriveling after 5 months in the refrigerated storage with gravity air movement (33.5 per cent) than in one where the air was moved by blower fans (21.5 per cent), because of the higher relative humidity held in the latter through the maintenance of higher average refrigerant temperature.

There was nearly twice as much shriveled fruit in the crates (36 per cent) as in the boxes or baskets (21.7 per cent), but in the latter containers, there were two and two-thirds more scalded fruits. Shriveled fruit appeared to a small extent (8 per cent) after 75 days in storage and nearly doubled in quantity by the hundredth day (14 per cent), after which the increase in number of fruits showing shrivel was moderate until the two-hundredth day (24 per cent). Scald appeared slightly in 100 days (4 per cent), nearly doubled in 150 days (7 per cent), and nearly doubled again (13 per cent) by the two-hundredth day.

The relative humidity of the storage air is of the greatest importance, since fruits should be stored in slatted crates to lessen the occurrence of scald. Forced-air cooling equipment maintaining a high refrigerant temperature answers these requirements. Good cultural and environmental conditions in the orchard, together with harvesting of the fruit at the proper stage, aid in delivering the best fruit possible out of the refrigerated storage to the markets.

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